

Vibration

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Vibration

Introduction

1 The vibration of ship structures is a wide subject for study and it is not the purpose of this chapter itself to provide an extended treatment of the phenomenon; this can be obtained from source material referenced in the text. The intended purpose of the chapter is to inform the ship designer of:

- a. The need to predict and assess the vibration behaviour of a new ship design.
- b. The methods which are available for the prediction and the criteria by which the calculated vibration may be assessed.
- c. The methods which are available for measuring the vibration behaviour of the completed ship.
- d. The methods of diagnosis and approach to solutions of vibration problems revealed during the trials of the completed ship.

2 The need to establish the vibration characteristics of a ship has, in recent years, assumed a greater priority in the ship design procedure as structures have become lighter because, firstly, a number of ships have suffered structural failure due to fatigue damage resulting from dynamic stresses, secondly, excessive vibration commonly results in damage to, or malfunction of, ships' equipment and, thirdly, ships' personnel subjected to excessive vibration experience discomfort and their proficiency in carrying out tasks is reduced due to fatigue. However, whether or not vibration is judged to be excessive depends upon the criterion to be applied, that is, fatigue damage, discomfort or some other malfunction. Appropriate limits for the various problems are discussed later in this chapter.

3 Long experience, presented for example by Ward (1983), has shown that the retrospective treatment of major vibration problems usually involves high cost and rarely provides a complete solution. Consequently, it is important that vibration minimisation should be considered as an integral part of the ship design process. In the initial stages of design, decisions are made on power, displacement, hull form and general arrangement, amongst other aspects, which dictate the fundamental vibration levels. At this stage a careful estimate and assessment of the vibration response might demonstrate that the basic ship design is likely to be clear of vibration problems. Or, perhaps more importantly, the study might reveal major, potential vibration problems which

will require early design modifications for their avoidance. Whatever the results at the early stages, however, as the design progresses the evaluation of the vibration aspects of the design needs to be a continuing process.

Basic Concepts

4 The design method for a structure which is intended to be free from excessive vibration must initially consider the vibratory forces to which the structure will be subjected. If these forces have a relatively high magnitude the structure will need to be designed with great care to ensure that its response to the forces will not interfere with its function. Provided that some control over the force characteristics can be exercised the design problem becomes easier. The constraints on the structural design, imposed by the necessity to avoid vibration problems, are then reduced and allow the structural properties to be dictated by the primary functions of the structure in resisting static and quasi-static loads.

5 Since most machines, both reciprocating and rotating, will produce some vibratory force it is obvious that there is a multitude of vibration excitation sources in the great majority of ships. In considering the ship as a whole structure, which is the case in the early stages of design, only three sources are of major importance:

- a. The propulsor.
- b. The main propulsion machinery.
- c. The sea.

The forces due to these sources are discussed in Chapter 3 and Chapter 4 and the evaluation of the dynamic response of a ship's structure to these vibratory forces generally involves two states of vibration, namely resonant vibration and forced vibration.

Resonant Vibration

6 When the natural frequency of a structure (that is the frequency at which the structure will vibrate freely when released after being displaced from rest) coincides with the frequency of some applied vibratory force a condition of resonance exists and the structure will vibrate at large amplitudes (which, in the absence of a damping mechanism, become infinite). Clearly it is desirable that a ship should not be subject to resonant vibration at the principal operational speeds. In some circumstances, however, resonant vibration cannot be easily avoided and it is then necessary to estimate the vibration amplitudes which will be excited at resonance. In order to do this the designer should be aware of the basic theory relating to the phenomena of natural frequency and damping of structures.

7 A very simple illustration of the theory involved can be obtained by considering a simple cantilever consisting of a single mass and a single spring stiffness shown in *Figure 14.1*. The mass is assumed to move in the transverse direction only and hence a single coordinate x is required to

define its position. The mathematical model to describe this situation falls in the category of a single degree of freedom system.

- 8 Applying Newton's law of motion, that is force = mass $\times$ acceleration, one can write

$$m\ddot{x} = -F = -kx$$

or

$$(14.1) \quad m\ddot{x} + kx = 0$$

in which m is the mass and k is the stiffness of the structure.

- 9 A general solution to equation 14.1 is

$$(14.2) \quad x = A_1 \cos \omega_o t + A_2 \sin \omega_o t$$

in which $\omega_o^2 = k/m$ and A_1 and A_2 are arbitrary constants to be determined from the initial conditions of motion. For example, if $x = 0$ when $t = 0$ then $A_1 = 0$ and $x = A_2 \sin \omega_o t$ which is clearly the equation of a sine wave with an amplitude of displacement A_2 . The motion defined by equation 14.2 is called Simple Harmonic Motion (SHM) which contains a single resonant frequency ω_o .

- 10 In the literature, for example Timoshenko and Young, 'Vibration Problems in Engineering', it will be found that vibration levels are expressed variously in terms of displacement, velocity or acceleration. It can be seen that for a single frequency sine wave that a simple relationship exists between these quantities, namely

$$(14.3a) \quad \text{velocity} \quad \dot{x} = A_2 \omega_o \cos \omega_o t$$

$$(14.3b) \quad \text{acceleration} \quad \ddot{x} = -A_2 \omega_o^2 \sin \omega_o t$$

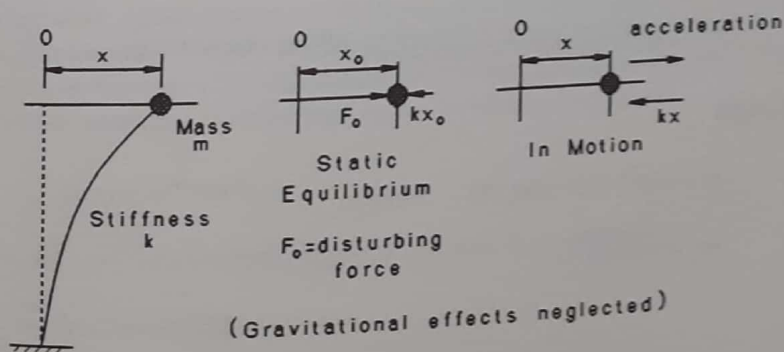


Figure 14.1 Single Degree of Freedom - Free Vibration

The period of oscillation, T , is then given by

$$(14.4a) \quad T = 2\pi/\omega_o = 2\pi \sqrt{m/k}$$

and thus depends on magnitude of the mass (m) and the spring constant (k) and is independent of amplitude. The natural frequency, f , is given by

$$(14.4b) \quad f = 1/T = \frac{1}{2\pi} \sqrt{k/m}$$

The natural frequency of a structure therefore is increased by increasing the stiffness or by reducing the mass.

Damping and Forced Vibration

11 The above is an ideal situation in which no energy is absorbed by the system and the amplitude of the oscillating displacement remains constant. In practice all structures in motion dissipate energy and the amplitude decays from the initial value. The rates of absorption of energy and the consequential amplitude decay depend on a property of the structure called damping. Energy absorption occurs due to hysteresis losses within the structural material itself, due to frictional effects where these exist and also due to dissipation in the surrounding fluid when the structure is immersed.

12 Linear viscous damping has the characteristic that the damping force is linearly proportional to the instantaneous velocity, that is, damping force = $C\dot{x}$ where C is a constant damping coefficient. The equation of motion of a single degree system including viscous damping and subject to a steady state harmonic force ($P_o \cos \omega t$) can then be written:

$$(14.5) \quad m\ddot{x} + C\dot{x} + kx = P_o \cos \omega t$$

Equation 14.5 is an extension to equation 14.1 including the damping term and with an exciting force of constant amplitude P_o .

13 The solution to the equation can be expressed as:

$$(14.6) \quad x = \frac{1}{\sqrt{\{(1 - \omega^2/\omega_o^2)^2 + 4v^2\omega^2/\omega_o^2\}}} \{P_o/k\} \cos (\omega t - \phi)$$

where phase angle

$$(14.6a) \quad \phi = \tan^{-1} \{(2v\omega/\omega_o)/(1 - \omega^2/\omega_o^2)\}$$

$$(14.6b) \quad \omega_o = \sqrt{k/m}$$

and

$$(14.6c) \quad v = C/(2m\omega_o)$$

When $\nu = 1$, that is when $C = 2m\omega_0$, the damping present in the system under conditions of free vibration just results in a non-oscillatory motion. The corresponding value of C is known as the critical damping coefficient C_c .

14 It is common to find in the published literature that damping is expressed as the ratio of the damping coefficient C to C_c , that is, damping ratio $\nu = C/C_c$.

Dynamic Magnification Factor

15 The displacement x that would occur under the application of a static force P_0 is P_0/k . From inspection of equation 14.6 therefore, it is seen that the factor K_m given by equation 14.7 acts as a dynamic magnification factor:

$$(14.7) \quad K_m = \frac{1}{\sqrt{\{(1 - \omega^2/\omega_0^2)^2 + 4\nu^2\omega^2/\omega_0^2\}}}$$

If damping is small, that is for $\nu \ll 1$, the maximum value of K_m occurs when $\omega = \omega_0$. This value of K_m has the approximate value of $1/2\nu$.

16 Equation 14.7 may be plotted to give a dimensionless measure of the amplitude and phase angle of the response as a function of the frequency ratio and is illustrated in *Figure 14.2*. This shows that if the forcing frequency is less than the natural frequency ($\omega/\omega_0 < 1$) the response tends toward the static response and the displacement is controlled by the stiffness of the spring and not by the mass or damping.

17 If the forcing frequency is near or equal to the natural frequency of the structure the response can be much greater than the static deflection, particularly if the damping is low. This coincidence of forcing frequency with the natural frequency is called resonance and in the resonant zone the amplitude is controlled by damping. For high forcing frequency, the response is controlled by mass rather than stiffness or damping of the structure. All these effects are shown in simplified form in *Figure 14.3*.

Logarithmic Decrement

18 An alternative quantification of damping can be expressed by considering that a displacement amplitude of a single degree of freedom system in free vibration will decrease logarithmically with time, the rate of decrease being a function of the damping. The ratio of the amplitudes of two successive displacement peaks, x_1 and x_2 , is given by:

$$(14.8) \quad x_1/x_2 = e^{2\pi\nu(\omega_0/\Omega)}$$

where $\Omega = \omega_0 \sqrt{1 - \nu^2}$ and the logarithmic decrement, δ , is defined by

$$(14.8a) \quad \delta = 2\pi\nu (\omega_0/\Omega)$$

when $\nu \ll 1$, $\omega \approx \Omega_0$ and $\delta \approx 2\pi\nu$.

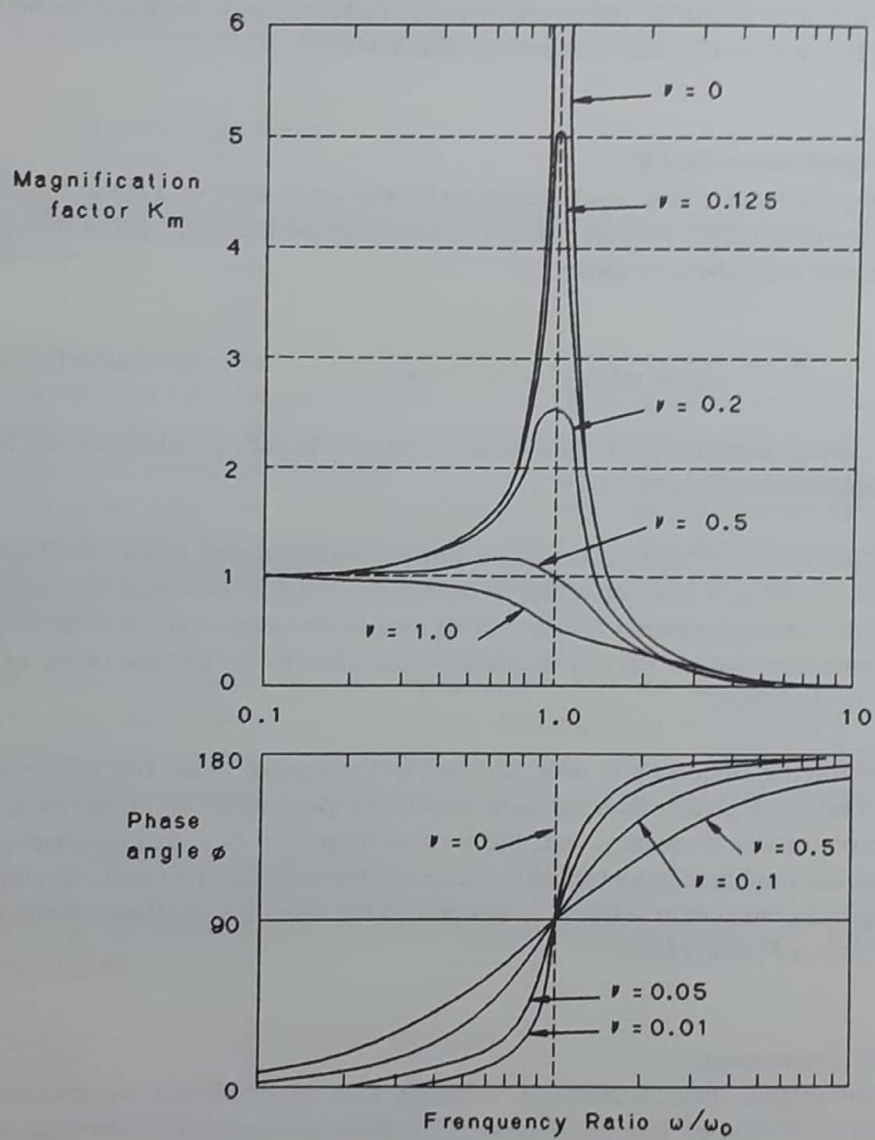


Figure 14.2 Variation of Magnification Factor and Phase Angle with Frequency Ratio

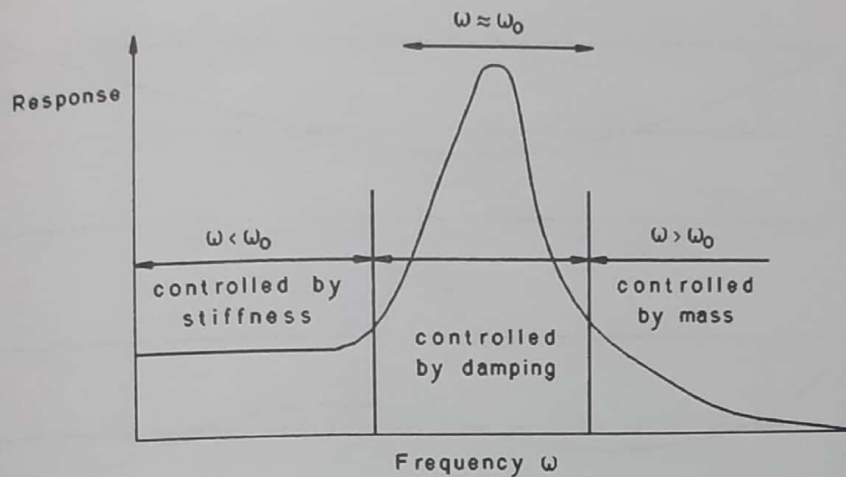


Figure 14.3 Frequency Response Curve

Application to Hull Girder Vibration

19 A ship's hull is a girder of a box-like construction and can be represented mathematically as a free-free elastic beam in which the distribution of weight and the structural properties, such as areas and moment of areas of cross sections, vary from one end of the ship to the other in a somewhat arbitrary manner (see, for example, Todd (1961) and Chalmers (1988(a))). Unlike the simple single degree of freedom system described above, the ship's hull girder can exhibit many modes of vibration. The simpler modes are usually assumed to be two dimensional, when vibration is restricted to vertical, athwartships or torsional motion and denoted according to the number of nodes (the positions of zero displacement) distributed along its length as illustrated in Figure 14.4. The higher modes of vibration, say those having more than six nodes in the vertical direction or four nodes in the athwartship direction, tend to be three dimensional as they couple with modes in other planes and cannot be simply described. The prediction of these higher modes requires extended mathematical modelling but fortunately the simple modes are usually those of more consequence in vibration design.

20 If the girder is subjected to periodic disturbing forces at frequencies close to any of these natural frequencies, resonant vibration conditions may occur and the amplitudes of vibration may exceed the limiting values set by the certifying or regulating authority.

21 Significant levels of vibration of the ship hull girder may result from vibratory forces even if the forcing frequency is well removed from the natural frequencies. This forced vibration is of concern, however, only when the exciting forces are relatively high, generally when the propulsor and main machinery are delivering high powers, and in most cases the estimated levels of forced vibration tend to be not excessive. Nevertheless, it is advisable to carry out such as estimate in order to establish the minimum levels to which the main structure of the ship will be subject at high speeds or powers. It is then possible to estimate an amplification of this vibration level resulting from resonance of local structures such as masts and seatings.

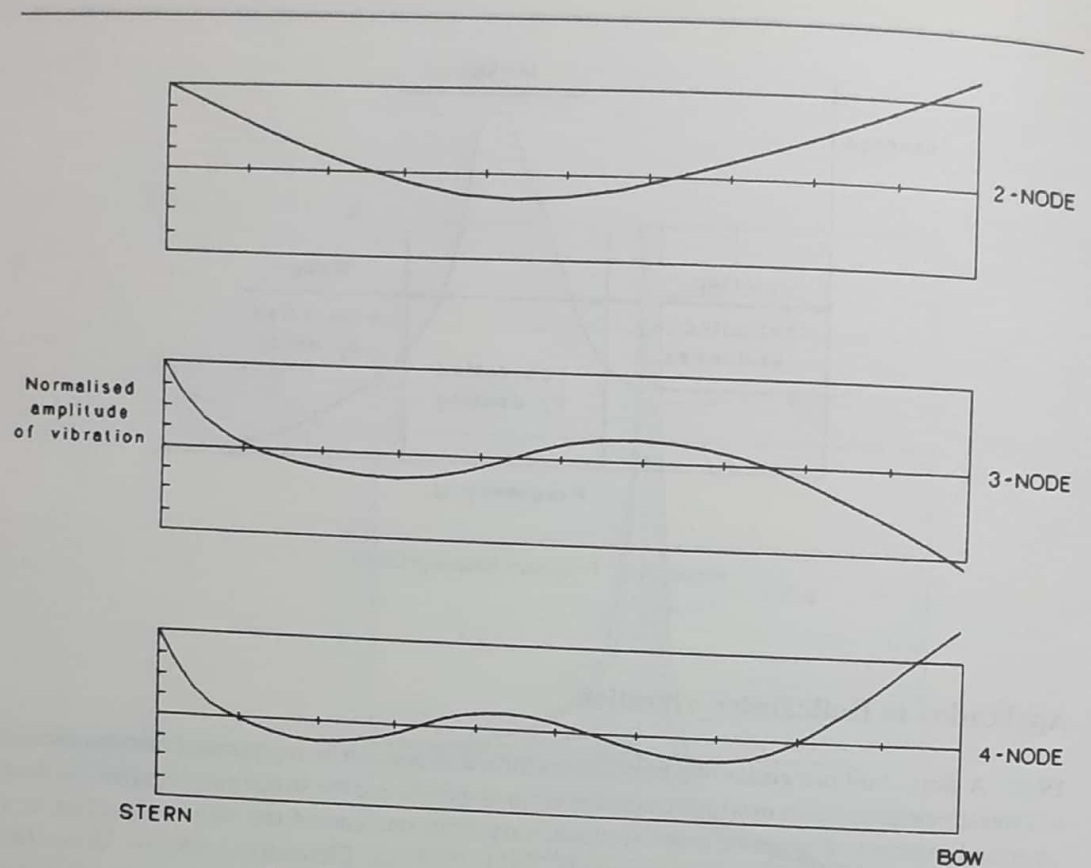


Figure 14.4 Typical Modes of Vibration of an Hull Girder

Propulsor and Shafting Forces

22 In ships without large reciprocating machinery the principal source of vibration is usually the propeller or propulsor. The oscillatory forces produced by the propeller arise from two separate phenomena. The most important are those induced by the operation of the propeller in a non-uniform flow field and are hydrodynamic in origin. Forces resulting from the mechanical unbalance of the propeller and shafting system also exist but these are usually, but not always, of less significance.

Hydrodynamic Forces

23 A propeller operating in the flow behind a ship generates oscillatory forces, as shown in Figure 14.5, which reflect the variations in torque and thrust reactions on the blades as they sweep through the changing flow velocities in the wake. The resultant forces and moments act through the shafting, the thrust block and the shaft bearings. The major frequency component of these forces is equal to the shaft rotational speed, the **shaft rate**, multiplied by the number of propeller blades and is referred as the **blade rate**. Generally these forces are of no great concern unless their frequency coincides with the natural frequency of some mode of the shafting system or of the ship's structure.

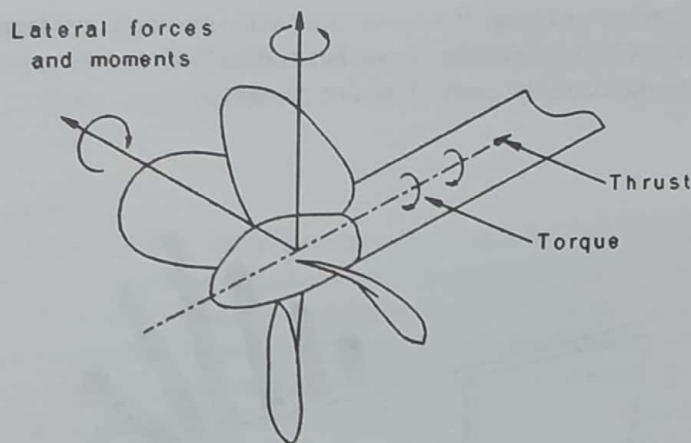


Figure 14.5 Oscillatory Propeller Forces and Moments

24 The amplitude of axial and torsional vibration of the propulsion train is largely independent of the ship structure and should be estimated during the design of the machinery system. The magnitude of the axial and torsional vibratory forces induced by the propeller are usually presented as some percentage of the steady thrust and torque, based upon collated measured data. The amplitude of lateral vibration of the shafting is more difficult to predict because of the lack of knowledge of the propeller forces and the damping associated with this mode of vibration. It is usually accepted that it is sufficient to ensure that the calculated natural frequency of lateral vibration is well removed from the excitation frequency, that is, the blade rate.

25 In addition to the vibratory forces transmitted to the ship through the shaft bearings, the propeller also produces an oscillatory pressure field at the adjacent hull plating as shown in Figure 14.6. This oscillatory pressure has a frequency corresponding to blade rate and would exist even if the propeller were operating in a uniform flow field. Where propellers are operating in a good wake, as should usually be the case, the hydrodynamic forces produced by the propeller may not be of any great concern, but as they are related to the power delivered they may well be of consequence in ships having relatively high power/displacement ratios. The determination of the amplitude of the pressure impulses emanating from a propeller has been the subject of extensive research over the last 10 years or so and there is a wide literature on the subject (see, for example, the proceedings of the International Towing Tank Conference (1987)). Pressure amplitudes can now be estimated with some confidence either analytically or by model testing, but unfortunately the same confidence cannot be attached to the estimation of the forces transmitted through the shaft bearings. However, experience suggests that, for propellers operating in a good wake, bearing forces are of a similar magnitude to the forces on the hull due to pressure impulses.

26 In the early stages of the ship design it is unlikely that the estimation of propeller forces will have been the subject of extended calculation or model testing. In that circumstance it is suggested that a first estimate of the total vertical vibratory force (bearing forces plus pressure forces) can be obtained from data sheet 18.85 assuming that the propeller is of a conventional design (that is without high skew or unconventional pitch distribution) and taking z on the data

sheet to be the *total* number of blades if a twin screw ship is being considered, and the factor K relating to the rise of floor being taken also from data sheet 18.85. In general it is only the vertical force which is of importance in the early stages of design.

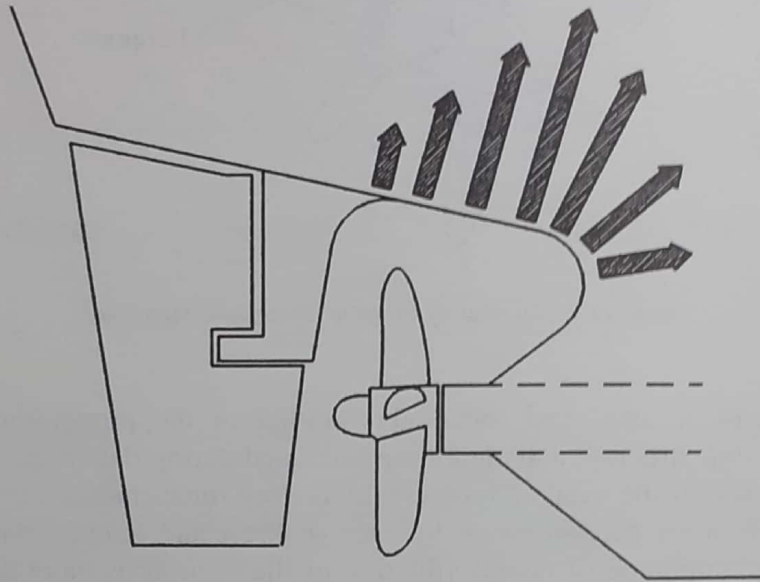


Figure 14.6 Area Subjected to Oscillatory Pressures

Propeller Cavitation

27 The description of the hydrodynamic propeller forces given above relates to a well designed propeller in a fairly uniform wake. If the propeller and wake combination is not satisfactory the propeller may cavitate excessively. Gross propeller cavitation, in the context of vibration, can be catastrophic since the pressure impulses generated by the propeller become much larger and more complex. *Figure 14.7(a)* shows pressure pulses from a typical non-cavitating propeller and *Figure 14.7(b)* shows pressure pulses typical of a heavily cavitating propeller. When cavitation is well established the force produced by the pressure amplitude increases approximately proportional to the cube of the power as discussed by Willshare and Bond (1978). Roughly, an increase of speed of 20% (say 20 kts to 24 kts) will result in blade rate vibration amplitude increasing by a factor of five. In addition, the pressure impulses contain frequency components relating to multiples of blade rate, which are of a significant amplitude as discussed by Fitzsimmons and Odabasi (1978). For example, a ship having a five bladed propeller operating at 180 revolutions per minute (rpm) would be subjected to relatively high vibratory forces at 15 Hz, 30 Hz, 60 Hz and so on. The resulting resonant response of stiffened panels and shell plating in the aft end region of the ship to these forces can result in fatigue fractures occurring within a fairly short period of time.

28 It is the task of hydrodynamicists and propeller designers to ensure that the arrangement of the propeller and hull configuration is such as to avoid the occurrence of excessive cavitation. The design of the structure is not a related problem except insofar as the hull shape at the after end and

the position, alignment and profile of bossings or shaft brackets are an integral part of the configuration and cannot be modified to facilitate structural design or production without compromising the hydrodynamic balance.

Mechanical Unbalance

29 Generally, careful attention is paid to the mechanical balance of propellers and shafting systems. If the balancing is carried out in accordance with normal limits, then the corresponding vibratory forces which have a frequency corresponding to the propeller rpm will not be large enough to induced unacceptable vibration of the ship's hull even under resonant conditions. To take a typical case a propeller and shafting weighing 25 tonnes should be balanced to have a remaining out-of-balance couple of about 60 Nm. At a speed of 170 rpm corresponding to the natural frequency of the 3-node vertical mode of a 3600 tonne displacement vessel, the vibratory force produced by the unbalance will be approximately 0.19 tonnes. This would result in a vibration velocity amplitude of about 2.5 mm/s which can be judged acceptable (see criteria and limits following, paragraphs 102 et seq).

30 If the designer is given the balancing details of a propeller, the corresponding vibratory force can be easily calculated and used to estimate the corresponding vibration amplitude of the hull. However, if a propeller is damaged in service both the mechanical and hydrodynamic balance will be disturbed and excessive vibration may result. A particular problem arises in the case of controllable pitch propellers which may have one or more damaged blades replaced without paying due attention to the consequent balance characteristics. Excessive vibration at a frequency corresponding to the shaft rate has been attributed to this cause in more than one ship.

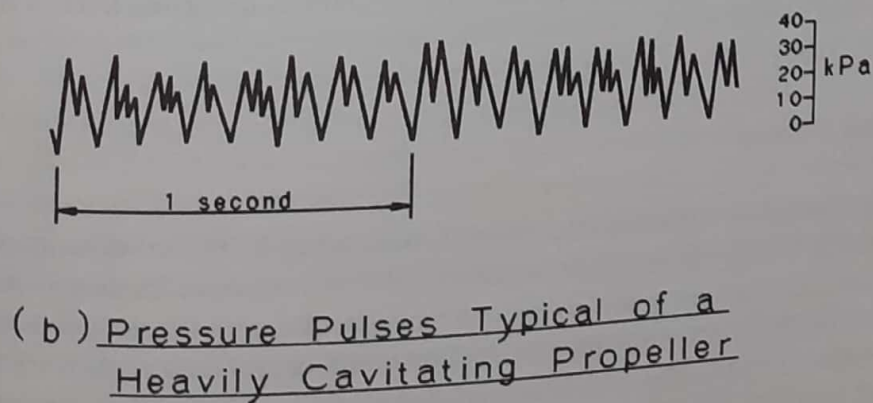
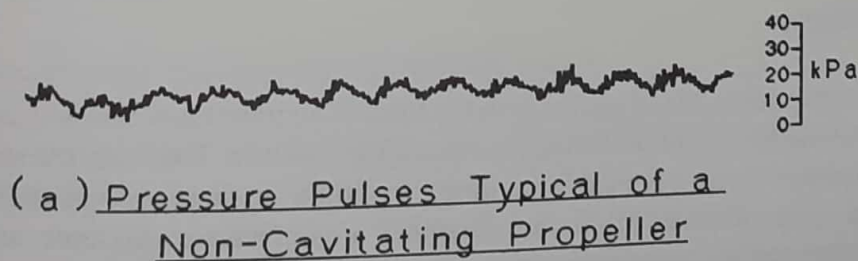


Figure 14.7

Machinery Forces

31 Rotating machinery such as turbines, electrical motors, generators and fans are not an important source of ship vibration except where underwater radiated noise is of concern. Any significant ship vibration induced by rotating machinery generally implies that the machine is not running satisfactorily. A good measure of vibration severity in these machines is the vibration velocity which should range from 0.125 mm/s peak velocity or 20 μm peak to peak displacement for very good condition to 16 mm/s peak velocity or 300 μm peak to peak displacement for unsatisfactory conditions. A frequent cause of vibration in purely rotating machines is unbalance which may change during the life of the machine because of damage, fouling, thermal distortion or wear. In some machines there may be fluid dynamic or electrical unbalance in addition to the mass unbalance. The accepted values of unbalance are usually expressed in N.m as a function of the maximum speed of the rotor.

32 Medium and high speed diesel engines are usually on anti-vibration mountings in warships and as such have little effect on the vibration of the ship. Care should be taken, however, to ensure that anti-vibration mountings do not result in amplification of vibration resulting from propeller excited structural responses.

33 Low speed diesel engines, particularly those having four or five cylinders can cause large, inherent vibratory forces, but such prime movers are rarely used in warships. An extended treatment of the forces generated by low speed diesel engines is given in the BSRA Ship Design Manual - Vibration (1981).

Gun Firing Forces

34 The excitation force due to gunfire is in the form of an impulse or series of impulses which can excite natural modes of vibration of the surrounding structure. The loads induced by guns are discussed in Chapter 4 where a typical force/time curve for a single shot is shown. Once the load-time curve has been obtained from the weapon's manufacturer, dynamic analysis of the supporting structure can be carried out using mode superposition or time integration methods and can be solved by a general purpose finite element program as discussed later in this chapter.

Sea Excitation Forces

35 Another source of vibration excitation is slamming, a detailed description of which is given in Chapter 3. Whipping response of the ship due to impulsive loading such as slamming, bow flare impact and shipping green water, can cause buckling of the deck or bottom structure. Measurements conducted onboard ships have shown that continuous vertical vibration can also create additional bending stresses which may be significant for fatigue considerations and that the stresses caused by whipping vibrations can be of the same order of magnitude as the wave bending stresses (see Chapters 3 and 13).

36 Springing is a continuous and steady vibration which arises from an unsteady hydrodynamic pressure field acting on the hull of the ship and is a result of coincidence between the natural frequency of the hull and the wave encounter frequency. Unless the hull has a very low bending stiffness due to large cut-outs in the structure of a high length/depth ratio, its natural frequency will relate to the higher frequency part of the sea spectrum where there is a limited energy content, and so springing responses are rarely of significance in warships. In general, the designer has limited influence on the response of the ship to sea excitation but might assess the position of sensitive equipment in relation to lower hull vibration modes as discussed by Chalmers (1988(b)).

Main Hull Vibration Response

37 The natural frequencies of the hull girder are a function of the ship's shape, mass and stiffness. These features are dictated by the displacement and strength requirements and, once built, the natural frequencies of the hull cannot be significantly altered by adding material to it (see Chalmers (1988(b))). Nor is it generally practicable in the design stages to alter the hull girder structurally to avoid resonant hull vibration. Instead, attention has to be given to the sources of the periodic exciting forces both in terms of their magnitudes and their frequencies.

38 Generally, the level of main hull vibration in any ship will depend upon:

- a. The relationship between the frequency of the forces and the natural frequencies of the hull girder.
- b. The effective magnitude of the exciting forces.
- c. The mode shape and hence the amplitude at both the excitation and response locations.
- d. The degree of damping present.

The designer is therefore faced with two approaches to ensure the avoidance of excessive main hull vibration. At a very early stage in the design process he must either arrange that the frequencies of the exciting forces do not coincide with any natural frequency of the hull, or ensure that the magnitudes of the forces are not sufficiently great so as to produce unacceptable vibration levels. In the following paragraphs both semi-empirical and analytical procedures for estimating the natural frequencies of the hull and the vibration amplitudes at resonance to known or estimated excitation forces from the main machinery and propeller, are briefly outlined.

39 The empirical methods described here for predicting the hull vibration characteristics have been developed from accumulated data obtained from sea trial measurements on many ships and are summarised in the BSRA Design Manual - Vibration (1981). This does not however include warship data which is not openly published.

Semi-Empirical Methods

40 Two methods are available, one employing an extrapolation method using similar ship data, the other a graphical approach using basic ship parameters. Both methods are essentially an adaptation of the simple empirical relationships developed for merchant ships and presented in the BSRA Ship Design Manual - Vibration (1981) but using the collected warship data.

41 The most reliable method of estimating the natural frequencies is on a comparative basis using similar ship data. Provided the 'similar ship' requirements are fulfilled the estimated frequencies of the lowest modes of vibration may be expected to be within an accuracy of about 3%. The graphical approach is unlikely, at this stage, to provide estimates with an accuracy better than 10%. If further data is accumulated it is expected that the more extended level analyses, involving the detailed functions, will provide more accurate predictions.

42 The preferred method of estimating the natural frequencies (N) of a new design uses similar ship data involving the following relationship:

$$(14.9) \quad N / \sqrt{\frac{I}{\Delta_1 L^3}} \text{ (new design)} = N / \sqrt{\frac{I}{\Delta_1 L^3}} \text{ (similar ship)}$$

in which I is the second moment of area amidships, Δ_1 is the displacement including added mass and L is the length between perpendiculars. The relationship applies equally well to vertical and horizontal hull resonances provided that the appropriate values of Δ_1 and I are used. As stated above, an estimate made this way should be within about 3% of the measured value.

43 In the case of torsional hull frequencies the current state of the analysis techniques is such that there are some uncertainties in establishing an appropriate relationship. However, one relationship, proven to be reasonably reliable for merchant ships, is:

$$(14.10) \quad N / \sqrt{\frac{J}{\Delta (B^2 + D^2) L}} \text{ (new design)} = N / \sqrt{\frac{J}{\Delta (B^2 + D^2) L}} \text{ (similar ship)}$$

in which Δ is the displacement (without added mass), B is the breadth moulded, D is the depth moulded and J is the torsional constant. Whilst the accuracy of the estimate is not expected to be high, it is considered sufficient to indicate to a designer that this form of vibration could pose a potential problem.

44 To facilitate the selection of a suitable similar ship the measured natural frequency data available at present for merchant ships is summarised in a tabular form for vertical and horizontal-torsional vibration modes, together with all the relevant ship parameters, in the BSRA Ship Design Manual - Vibration (1981). For the purposes of illustration and to provide data for the principal vibration modes, those suitable for vertical vibration of warships are reproduced in data sheet 18.86. The midship sections property values, I_V (I vertical amidships), I_H (I horizontal amidships) and J are derived using standard methods, for example as described in Bishop et al (1986). The total virtual added masses for vertical and horizontal vibration (denoted by Δ_{1V} and

Δ_{1H} respectively) have been calculated using the same expressions as used for merchant ships, namely:

$$(14.11a) \quad \Delta_{1V} = \Delta (1.2 + B/3T)$$

and

$$(14.11b) \quad \Delta_{1H} = \Delta + 0.574 LT^2$$

for vertical and horizontal vibration respectively. Δ (in **tonnes**), B and L are as defined above and T is the mean draught (all in **metres**).

45 If the designer in the earliest stages of the design process is not able to identify a 'similar' ship or has not yet determined all the necessary parameters for that method it is still possible to estimate approximately the hull frequencies using one of a number of graphs. However, up to now satisfactory relationships have been developed only for the lower vertical and horizontal frequencies; the most reliable graph for estimating the 2-node mode of vertical vibration is shown plotted in data sheet 18.87, but alternative graphs using other base functions are available in the BSRA Ship Design Manual - Vibration (1981). In data sheet 18.87 both merchant ship and warship data are plotted against the parameter

$$\sqrt{\frac{I_V}{\Delta_{1V} L^3}}$$

with units in **tonnes** and **metres**. The mean lines on each plot have been calculated from least-squares fit.

46 The advantage of the graphical approach is that the designer is able to carry out a rapid frequency estimate as design information becomes available by simply entering the graphs at the appropriate base functions. The relationships between the 2-node mode and the natural frequencies of higher modes are presented in a graphical form in the BSRA Ship Design Manual - Vibration (1981). Note that no graphs are presented for estimating torsional natural frequencies since these have been found to be inconsistent using the data currently available.

Superstructure Correction Factor

47 The hull natural frequencies are influenced by the extent of any superstructure and it must be taken into account if reliable frequency estimates are required. The method used to take into account a warship's superstructure is to multiply the hull only vertical section inertia value (I_V) by a factor of k_{ss} given by the following expression to give a modified value of I_V for use with either the similar ship or graphical methods:

$$(14.12) \quad k_{ss} = 1 + r (I_{100}/I_V - 1)$$

where r is the ratio of the mean length of the superstructure to the waterline length and I_{100} is the second moment of area amidships with the superstructure fully effective, as described in Chapter 8.

Hull Resonant Amplitudes

48 It is inevitable that one or more of the main hull natural frequencies will be excited into resonance by, for example, reciprocating machinery or by the propeller at some point in the ship's speed range. The designer should seek to avoid this condition arising at any speed at which the ship will be operating for relatively long periods and especially at the higher speeds where excitation forces tend to be relatively high. In general a 20% difference between natural and excitation frequencies is recommended but this could be reduced to a lesser percentage, say 10%, provided that the estimate of the natural frequency is based on reliable data. If the avoidance of resonance is impractical then it is necessary to estimate the vibration amplitudes that will occur in such circumstances and compare them with established vibration limits.

49 In order to carry out this estimate it is necessary to have information on the damping characteristics, effective vibrating masses, and deflection profiles for the various hull modes and, of course, the magnitude and frequency of the forces. In the case of diesel engines which are normally the major excitation source in merchant ships, the forces can be calculated accurately and are usually supplied by the engine manufacturer. In principle, the following method can be used to calculate the resonant vibration amplitudes resulting from any force.

50 The general expression for the velocity amplitude at any point along the ship, for horizontal and vertical vibration, is given by Johnson et al (1961-62) as follows:

$$(14.13) \quad \dot{y}, \dot{z} = \{93660 \alpha_x K_m / (C_1 N \Delta_1)\} \sum F \alpha_F$$

where Δ_1 is the total virtual mass, N is the n^{th} natural frequency in cycles/min, K_m is the dynamic magnification factor (see equation 14.7), C_1 is the generalised mass coefficient, α_x is the normalised amplitude of vibration at coordinate x , and

$$(14.13a) \quad \sum F \alpha_F = Q \alpha_Q + M i_m / L$$

where α_Q is the normalised ordinate of the amplitude at the point of application of the force Q , i_m is the slope factor at the point of application of the couple M and L is the length overall. The method of calculating the values of these parameters are fully explained in Chapter V of the BSRA Ship Design Manual - Vibration (1981).

51 If there is a likelihood of the out-of-balance forces exciting a main hull resonance then the amplitude of vibration at the natural frequency can be derived by substituting the appropriate values of the parameters in equation 14.13. In applying the general case, care should be taken to ensure the signs of the forces and couples are consistent with the deflection and slope of the modal profile. Note that if the source of excitation is a diesel engine only couples are present.

52 There are three common ways of expressing the linear viscous damping associated with a vibrating structure. These are damping ratio (ν), dynamic magnification factor (K_m) and logarithmic decrement (δ) as discussed earlier in paragraphs 12 et seq. The use of K_m factor has the advantage that it directly reflects the vibration amplitude, that is the higher the value of K_m , the higher will be the amplitude of vibration. The assignment of appropriate dynamic magnification factors values to the modes of vibration of a ship's hull can pose a problem. Because of the

many physical mechanisms contributing to the damping of vibration in a ship under way there is not yet means of deriving K_m from the basic ship particulars, for example length, breadth and displacement. Access to data obtained for a similar ship is the most reliable source but such data is sparse.

53 Dynamic magnification factors for merchant ships are graphically presented in *Figure 14.8*. Shown superimposed on the merchant ship curves are data points collated from only four warship trial measurements. Although limited, the indications are that the mean damping line appears to be reasonably representative for steel hulled warships. As might be expected, the damping for FRP constructed ships has been found to be significantly greater, that is by a factor of four.

54 The coefficient C_1 in equation 14.13, the generalised mass coefficient, takes account of the effective vibrating mass for each mode of vibration and is dependent on the actual weight distribution and deflection profile. Whilst differences are likely to exist between merchant and warship mode profiles their similarity is sufficient, as shown in *Figure 14.9* for the two node vertical mode (other modes follow the same trends), to adopt merchant ship values of C_1 for the purpose of estimating warship vertical vibration amplitudes. These are:

2-node	3-node	4-node	5-node	6-node	7-node
0.092	0.080	0.075	0.093	0.110	0.125

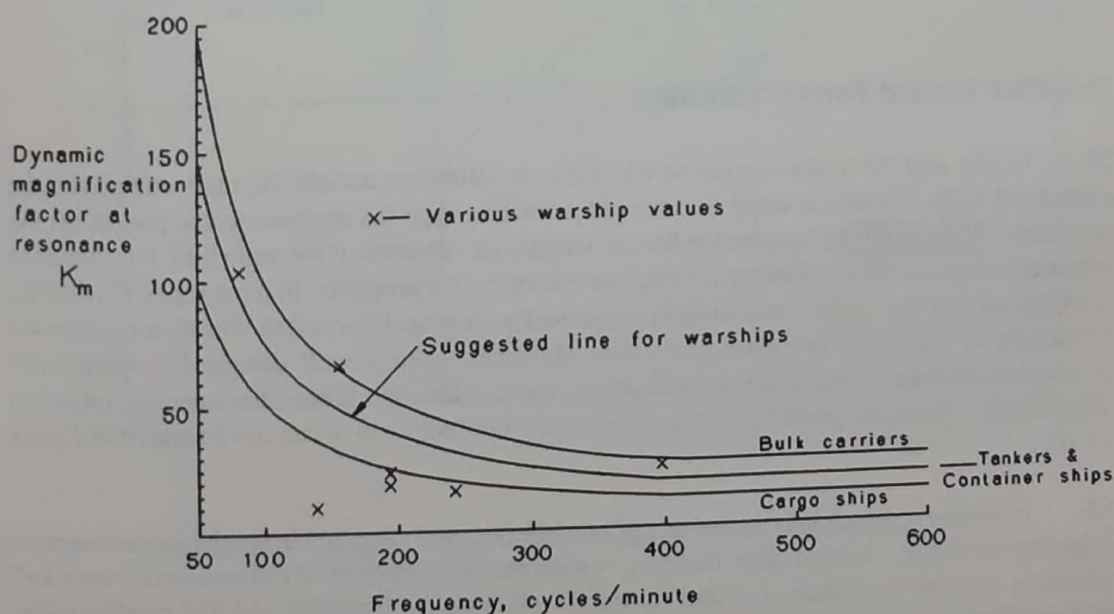


Figure 14.8 Damping Coefficients for Main Hull Vertical Modes of Vibration

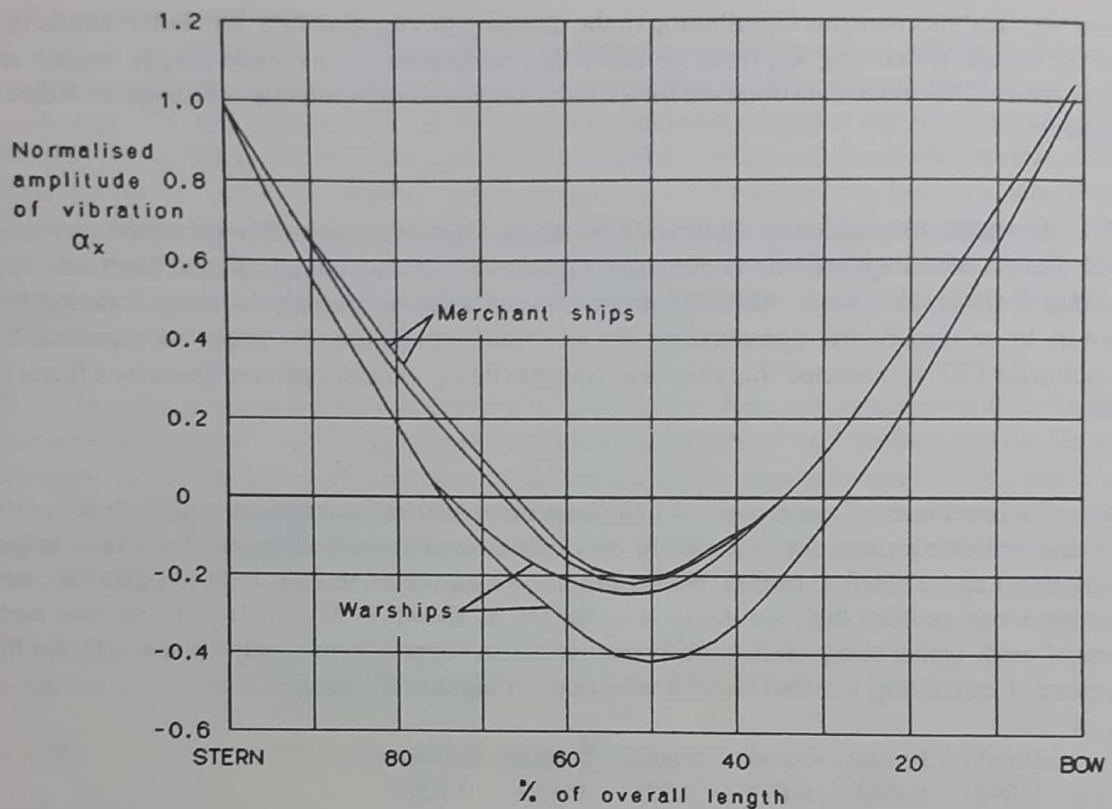


Figure 14.9 Typical 2-Node Vertical Vibration Mode Profiles

Propeller Excited Forced Vibration

55 In any ship the major sources of vibration excitation are usually the main engine and the propulsion train. Where the main engine is of a rotating type (as opposed to reciprocating) the propeller will probably be responsible for any significant vibration of the main hull. It is therefore important to assess the amplitudes of vibration excited by the propeller. In paragraphs 27 and 28 it is explained that the major consideration associated with propeller excited vibration must be the avoidance of excessive propeller cavitation. However, even a well designed propeller will produce oscillatory pressures on the hull above it and, where the power/displacement ratio of a ship is high, significant propeller excited vibration amplitudes can occur, as illustrated in *Figure 14.10*.

56 Extended research into the subject of propeller excited vibration in single-screw merchant ships has provided, amongst other things, a simple graphical method of estimating the main hull vibration velocity amplitude of a ship having a well designed propeller and hull configuration. Unfortunately, the same comprehensive measurement and calculation programmes have not been extended to twin-screw warships. However, on the basis that the power/displacement ratio is the most significant parameter, a first estimate of propeller excited amplitudes might be obtained by

using the information on single-screw merchant ships given in the BSRA Ship Design Manual - Vibration (1981), but for warships experience suggests that any such estimate should be modified in the light of the following knowledge:

- a. The vertical propeller excited vibration measured in the hull above the propellers of twin-screw ships is usually less than half of that measured in single-screw ships of the same power/displacement ratio.
- b. The design of propellers having low noise characteristics usually (but not always) results in lower levels of propeller excited hull vibration.

57 At a later stage in the design process, that is when propeller and hull geometry are defined, oscillatory pressures induced by the propeller can be calculated. Later still, in the case of most ships, model tests will provide supporting measured data. The vibratory forces, when calculated or measured, can then be used as input to finite element models for the calculation of hull vibration amplitude as discussed below.

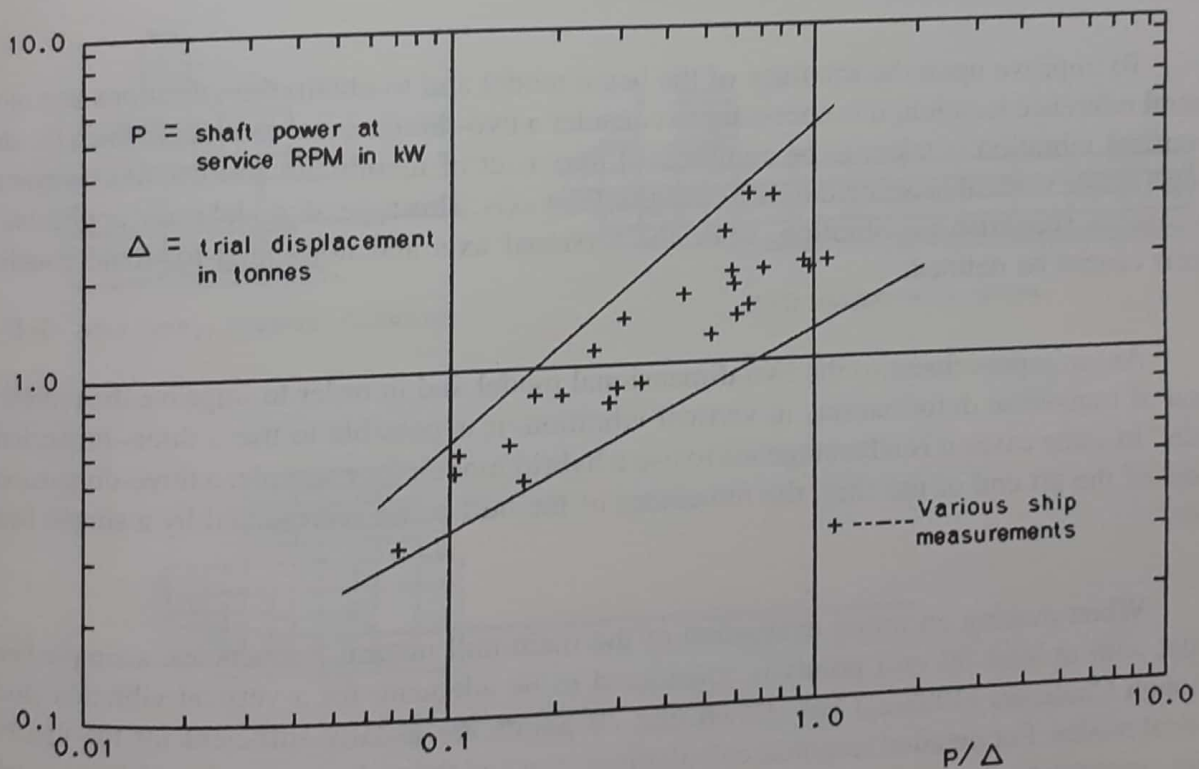


Figure 14.10 Propeller Excited Vertical Vibration Power/Displacement Relationship

Analytical Methods

Finite Element Analysis

58 The method of finite element analysis (FEA) provides a powerful design tool in circumstances where a less conventional ship is under design, and has the advantage that the model is based upon a structural definition rather than a mathematical model, and knowledge of the behaviour of individual structural components can be accounted for. Numerous dynamic response analyses of ship structures have been carried out and a brief description of various tasks involved in the method are mentioned here. It must be noted, however, that for sufficient data to be available the structural design must be at a relatively advanced stage.

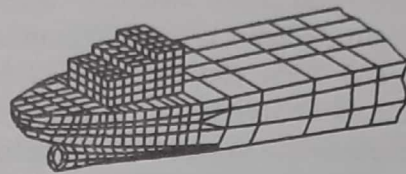
59 The several types of FEA models for the hull girder response fall into three main groups, beam models, two-dimensional models and three-dimensional models, and they, and several hybrid models, are illustrated in *Figure 14.11*. The beam model approach is the simplest finite element model with which to represent a ship structure and is therefore the obvious first one to consider. In this method the hull girder is represented as a union of a single line of finite beam elements along the main hull axis. Each element is assigned mass and stiffness properties derived from the ship's structure and the loading distribution at that section, together with an estimate of the added mass effect of the entrained water.

60 To improve upon the accuracy of the beam model and to obtain the vibrations at a more general reference location, it is necessary to consider a two-dimensional model in which the ship in vertical vibration is taken to be compressed into a set of membrane and bar finite elements along a single vertical plane on the ship's longitudinal axis. This type of model is not applicable to the case of athwartships vibration, since the torsional axis and horizontal-torsional coupling effects cannot be defined.

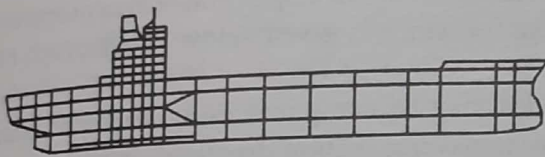
61 As an improvement to the two-dimensional model and in order to improve the representation of transverse deformations in vertical vibration, it is possible to use a three-dimensional model. In some cases it is advantageous to use a hybrid model, for example, a three-dimensional model of the aft end of the ship, the remainder of the hull being represented by a simple beam model.

62 When making an initial evaluation of the main hull natural frequencies, a simple beam model with at least 30 grid points is considered to be adequate for a vertical vibration study, although Chalmers (1988(a)) has shown that 20 points are usually sufficient for the first two vertical modes. For detailed response calculations, two and three-dimensional models are usually more appropriate.

63 The natural frequencies of the above models are best calculated using a program which commences at the lowest mode; a plot of the corresponding mode shapes is then obtained. It is advisable to consider first the validity of the lowest modes with reference to similar ships or empirical values based on basic ship parameters before proceeding to calculate higher modes so as to avoid considerable wasted effort.



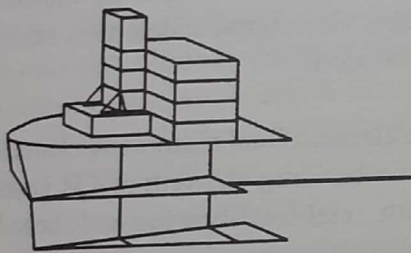
Full three-dimensional model
(part only)



Two dimensional model



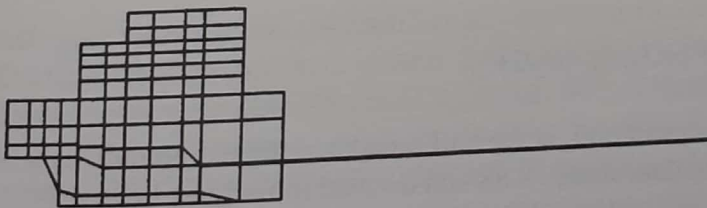
Beam element model



3-D afterbody, beams forward



2-D model with beams



2-D after body, beams forward

Figure 14.11 Examples of Finite Element Model Types

64 For a beam model of a ship in vertical vibration the parameters that are required for the input are total structural area, vertical shear area and vertical second moment of area which are calculated for several, say eight, transverse sections. The appropriate values for each structural parameter are then assigned to the 30 grid points based on interpolation from the eight values calculated. Also required for the beam elements is the longitudinal distribution of the total ship weight, including variable load. This total can be checked by comparison with the ship displacement. A detailed distribution of the weight may not be available, in which case it will be necessary to use an estimated distribution. It is important that the total weight is correct even if the distribution is not known with confidence. An example of such a calculation is presented by Chalmers (1988(b)).

65 The added mass of the entrained water is especially important in vertical vibration as it is of the same order as the ship mass. It is calculated from the sectional displacement area properties and the breadth and depth of each section, a formula for which is given below in paragraph 68.

66 The horizontal and torsional vibration modes may be computed using a similar beam model type representation. For this the distribution of transverse section structural properties such as horizontal shear area, horizontal second moment of area, torsional constant, vertical position of centre of gravity of section, position of shear centre are required. The additional mass information that is necessary includes mass, polar mass moment of inertia about centre of gravity and vertical centre of gravity of ship structure, outfit, machinery and cargo. Horizontal added mass and polar added mass moment of inertia of entrained water are also needed.

67 In the two-dimensional model finite elements are bar elements and membrane elements. A bar element is merely an axial load-bearing member, has no bending strength and is generally placed along an edge of a membrane element which has both axial-load-bearing and shear-load-bearing capability. The thickness of membrane element and cross sectional areas of bar elements are appropriately calculated. The loads including the added mass are also required.

68 The added mass per unit length (m_v) at a particular section may be calculated using equation 14.14

$$(14.14) \quad m_v = 0.5 \rho \pi J_M C_V (B_2/2)^2$$

where ρ is the density of seawater, B_2 is the half breadth at the required draught height, C_V is the Lewis shape factor given in data sheet 18.88 and derived initially by Lewis (1957). J_M is a three-dimensional correction factor, and based on the ship dimensions, a value applicable to all sections may be taken from equation 14.14a taken from Landweber and Macagno (1957)

$$(14.14a) \quad J_M = 1.016 - 1.58 B_2/L$$

69 In order to calculate added mass values at 30 different sections in a finite element model and also to have a more accurate representation of the required parameters, the calculations can be carried out using an appropriate program such as UCLMARS described by Bishop et al (1977) or VAS due to Catley (1977). These programs are usually based on Lewis form theory with a correction applied to account for the three-dimensional nature of flow. A simple two-parameter

Lewis fit is considered to be a sufficiently accurate representation of each section. The total added mass obtained from the program should be checked against an empirical estimate.

70 Typical times for the preparation of data for a beam model for vertical vibration would be of the order of three to six days depending mainly on the number of sections at which the structural properties of the hull girder are to be calculated. For an analysis of horizontal and torsional vibration, an additional one to four days and comparable computer effort would be then required. The time to complete the FE analysis could be reduced as the user became more experienced with the system and if processed data from the modelling of a similar ship were available. Consequently, it is difficult to be conclusive about the cost effectiveness of the three-dimensional model but it always requires a large computing capability and a large amount of data preparation. This tends to preclude its use in routine design work but may be essential when the designer is involved in ships of unconventional or advanced design. However, to analyse any structure which is asymmetric about the centreline, such as an aircraft carrier, a form of three-dimensional model is essential, as shown by Bishop et al (1986).

Mobility and Local Vibration Amplitude

71 It is important for the designer to know the local vibration amplitude in a ship's structure due to the various excitation forces arising from machinery and equipment, and to check whether it exceeds the limiting criteria. The calculation of mobility, which is the vibration velocity response of a structure to a unit exciting force, will enable this amplitude to be determined fairly readily. In other words, if the mobility is calculated for some point on a ship's structure then the vibration velocity excited at that point by some known force is immediately available.

72 There are two definitions of mobility; **point mobility** is the ratio of the vibration velocity at a point to the exciting force at the same point, while **transfer mobility** is the ratio between the velocity at one point and the force at a different point. Since the velocity and force need not be in phase, mobility is a complex quantity.

73 Many full-scale measurements of mobility have been carried out by organisations such as BMT CORTEC Ltd, mainly on merchant vessels, and it has been found that a comparison of the average mobilities measured at frequencies of less than 600 cycles/min (the propeller excited blade rate range) at the stern gland, suggests that the mobility is generally related to the displacement of the ship. The relationship derived is reproduced in data sheet 18.89. It is suggested that the empirical relationship given in the data sheet could be used in the early stages of a new design to assess average mobilities but, more importantly, should serve as a check for the mobilities calculated by FEA methods up to a frequency of 600 cycles/min.

74 The calculation of mobility by FEA methods for a particular ship is a relatively simple procedure, once the natural mode frequencies and mode shapes have been found as described above, and an estimate is available for the modal damping. One empirical approach for the estimation of damping which has been considered by Hylarides (1974), suggests that the actual damping in a particular mode can reasonably be taken as being proportional to the stiffness of the mode. The constant of proportionality (x) is dependent, to some extent, upon the degree of

idealisation of the ship structure. Suggested values of damping factor for three typical values of α are shown in data sheet 18.90.

75 Engine-excited vibration amplitudes may be derived from calculated mobilities or, more usually, directly by assigning absolute values of the unbalance forces and moments at the appropriate model grid points in the finite element model corresponding to the engine position. *Figure 14.12* compares the calculated and measured amplitudes for a products tanker excited by main-engine secondary unbalance couples. For convenience these are shown plotted on a linear basis.

76 Finite element models using any of the methods discussed in paragraphs 58 et seq may be employed to calculate the vibration response at each grid point of the model for excitation frequencies less than the highest calculated natural frequency of the model. Such a model may be used for both propeller-excited and engine-excited vibration calculations; however, in the latter case the flexibility of the seating arrangement should be included and spring elements are suitable for this purpose. The procedure briefly comprises the following:

- a. Select finite element model;
- b. Assign modal damping values (see data sheet 18.90);
- c. Assign excitation forces or point excitation for mobility;
- d. Select frequency increments for response calculations;
- e. Compute and assess responses.

An illustration of the use of FEA methods to determine the vibration velocity amplitudes of a ship excited by engine and propeller forces is given in Chapter V of BSRA Ship Design Manual - Vibration (1981).

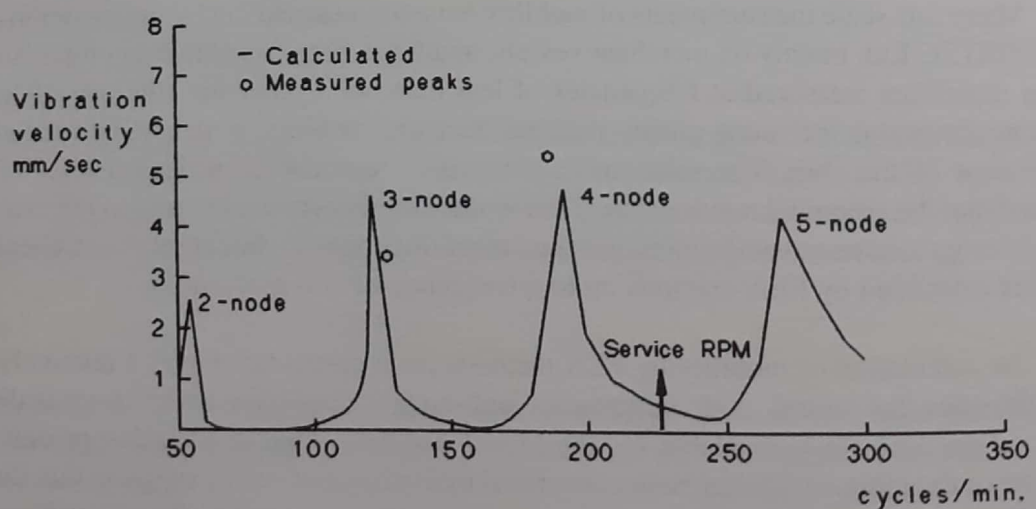


Figure 14.12 Typical Vertical Response at Stern Gland to Engine Secondary Unbalance

Vibration Response of Local Areas

77 In addition to ensuring that the vibration characteristics of the main hull girder are likely to be satisfactory, it is essential to ensure that local areas of structure such as masts and stiffened panels should not vibrate excessively.

Mast Vibration

78 Guidelines on the design of a mast on the basis of static strength has already been covered in Chapter 12 (including a discussion of margins) and the design is generally based on loads as defined in Chapters 3 and 4, that is to withstand the load effects due to combined effect of inertia load caused by wave action, wind and underwater shock. There are three types of mast in general use, the pole mast, the lattice mast and the plated mast. The advantages and disadvantages of these masts are also briefly mentioned in Chapter 12.

79 Apart from resisting the loads, one of the requirements of masts is that they should not be dynamically over-sensitive. In other words the frequencies and amplitudes of vibration should be such as not to interfere with sensor performance, and hence a check on frequency response is necessary in order to see whether they will be excited by hull motions or machinery. If it is seen that the natural frequency is near to the exciting frequency, then a resonant condition will occur and further checks on amplitudes will be needed in order to discover whether they are within acceptable limits.

80 There are a number of methods available which can be used at an early design stage to calculate the vibration behaviour of masts and other similar appendages. It is difficult to assess the accuracy of the calculation methods because of the limited amount of measured data, available either in the published or unpublished literature, on the natural frequencies, mode shapes and vibration amplitudes of appendages. In many cases the stiffness of the supporting platform is the principal feature controlling the natural frequency and in such circumstances any attempt to increase the natural frequency of the mast must take into account the influence of the stiffness of the supporting structure.

81 There are a number of approximate methods and formulae available in the literature, for example Clough and Penzien (1975) and Blevins (1979), which can be useful in estimating the first natural frequency of vibration of a mast, particularly if it can be approximated as a cantilever. One of the most valuable approaches involves the use of energy methods, the Rayleigh method for single degree of freedom systems and its extension Rayleigh-Ritz method for multi-degree of freedom systems. The basis of the energy method is that the sum of the kinetic energy of the system and its potential energy during a cycle of motion is constant. This produces, for single degree of freedom systems, the expression for the fundamental circular frequency of:

$$(14.15) \quad \omega^2 = \frac{EI \int_0^h (d^2y/dx^2)^2 dx}{\int_0^h \rho A y^2 dx + \sum m_i x_i^2}$$

in which EI is the flexural rigidity of the beam of length h , ρ is the material density, A is the area of cross section and m_i is the lumped mass for discrete items at distance x_i from the origin. Provided

that an expression for the deflected shape of beam $y = f(x)$ is known or assumed and the beam properties, together with its point loads, lumped mass m_i and distributed mass ρA are given, ω^2 can be obtained. The simplicity and accuracy of the method depends upon the choice of $f(x)$.

82 The deflected shape of a cantilever can be assumed to be of the form of part of a cosine function, that is:

$$(14.16) \quad y = y_o \{1 - \cos (\pi x/2h)\}$$

where y_o is the deflection of the unsupported end and the base is clamped. Such functions have wide validity for this type of problem but the question of base fixity should also be addressed, and if it known then a linear function such as $y = y_1 x/h$ may be added to equation 14.16 at the expense of some complexity in the solution.

83 Applying equation 14.15 to a uniform cantilever with a concentrated mass at its end, the fundamental natural frequency of the beam may be calculated by the following approximate formula:

$$(14.17) \quad f_m = 51.8 \sqrt{\{EI/(M + 0.23m) h^3\}}$$

where f_m is in cycles/min, M is the concentrated mass at the end of the cantilever of length h and m is mass of the cantilever itself, the expression in brackets being in consistent units.

84 Equation 14.17 is strictly valid for a cantilever structure having uniform flexural rigidity along its length. A mast type structure is generally of varying stiffness gradually decreasing from the bottom to the top of the structure. In this case careful judgement is required in the application of equation 14.17 for estimating the equivalent flexural rigidity of the beam. However, the method can give a useful first estimate of the natural frequency which should be within 20% of the true solution.

85 For a cantilever structure with varying stiffness and mass distribution equation 14.15 can be applied assuming a mode shape as given in equation 14.16 and suitable expressions for the variation of stiffness and mass with x . An expression for the fundamental natural frequency, similar to that given in equation 14.17, will then be obtained but the complexity of the solution is much increased.

86 Some of the limitations of such approximate formulae are:

- a. The boundary condition is limited to clamped for practical purposes; experience shows that the measured value of frequency on a real mast is 20 to 30% lower than the calculated value.
- b. The standard formulae cover a limited range of structural shapes, stiffness and mass distribution.
- c. They can only reliably predict the first natural frequency.

87 The finite element method can be used when no other analytical solution is applicable to the type of mast concerned. The potential advantages of the finite element method are:

- a. It can deal with any complexity of stiffness, shape and mass distribution.
- b. It can predict a large number of natural modes of vibration.
- c. The boundary conditions can be specified to represent the flexibility of the supporting structure.
- d. The influence of the vibration of the structure on which the mast (or any other appendage) is mounted may be accounted for by attaching the appendage model to a model of the whole structure.

88 The accuracy of the finite element calculation is very much dependent on the modelling technique used, on choosing suitable element types, knowing the boundary constraints and on modelling the stiffness and mass distribution of the structure with sufficient accuracy. The pole mast should be analysed as a series of beam elements. The lattice mast should be treated either as plane frame or space frame structure depending on its arrangement, and the plated mast can be treated as a series of beam elements. The values of second moment of area, torsion constant and shear area of the elements are required and considerable care should be taken in the evaluation of these parameters. The shear deflection in particular is appreciable in plated structures. Most FE programs can be used for prediction of natural frequencies and mode shape of mast type structures and typical mode shapes of a cantilever lattice mast are shown in *Figure 14.13*.

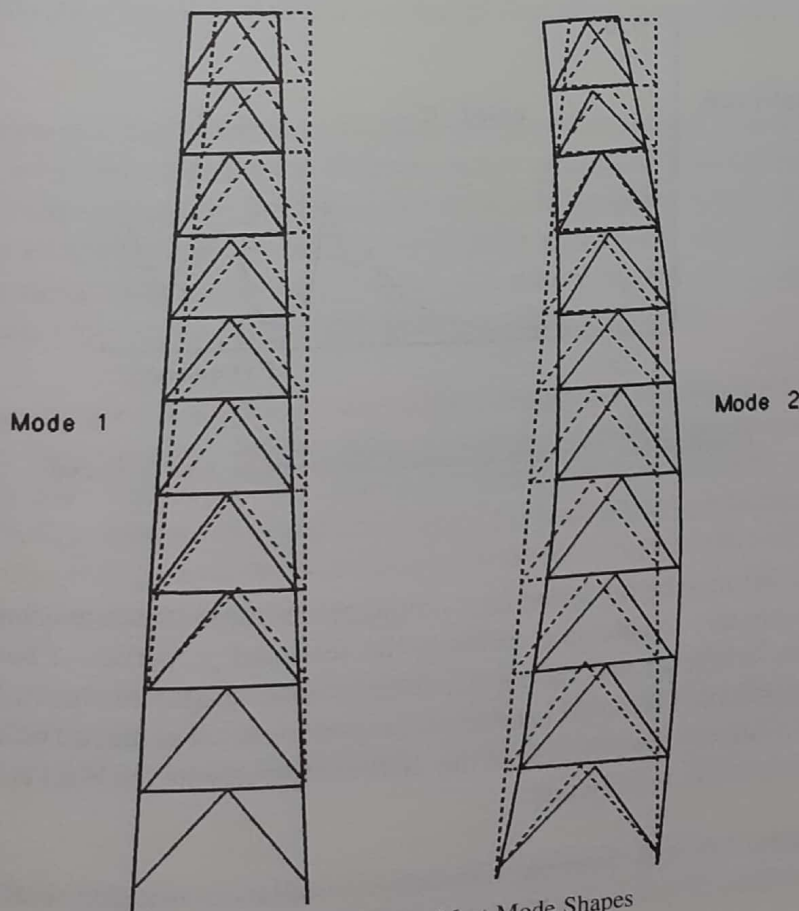


Figure 14.13 Lattice Mast Mode Shapes

Shaft Brackets

89 It is advisable to check the vibration response characteristics of shaft brackets for any resonant condition in relation to the frequencies of propeller vibration excitation. Typical response characteristics for an A bracket measured on a North Sea ferry are shown in *Figure 14.14*.

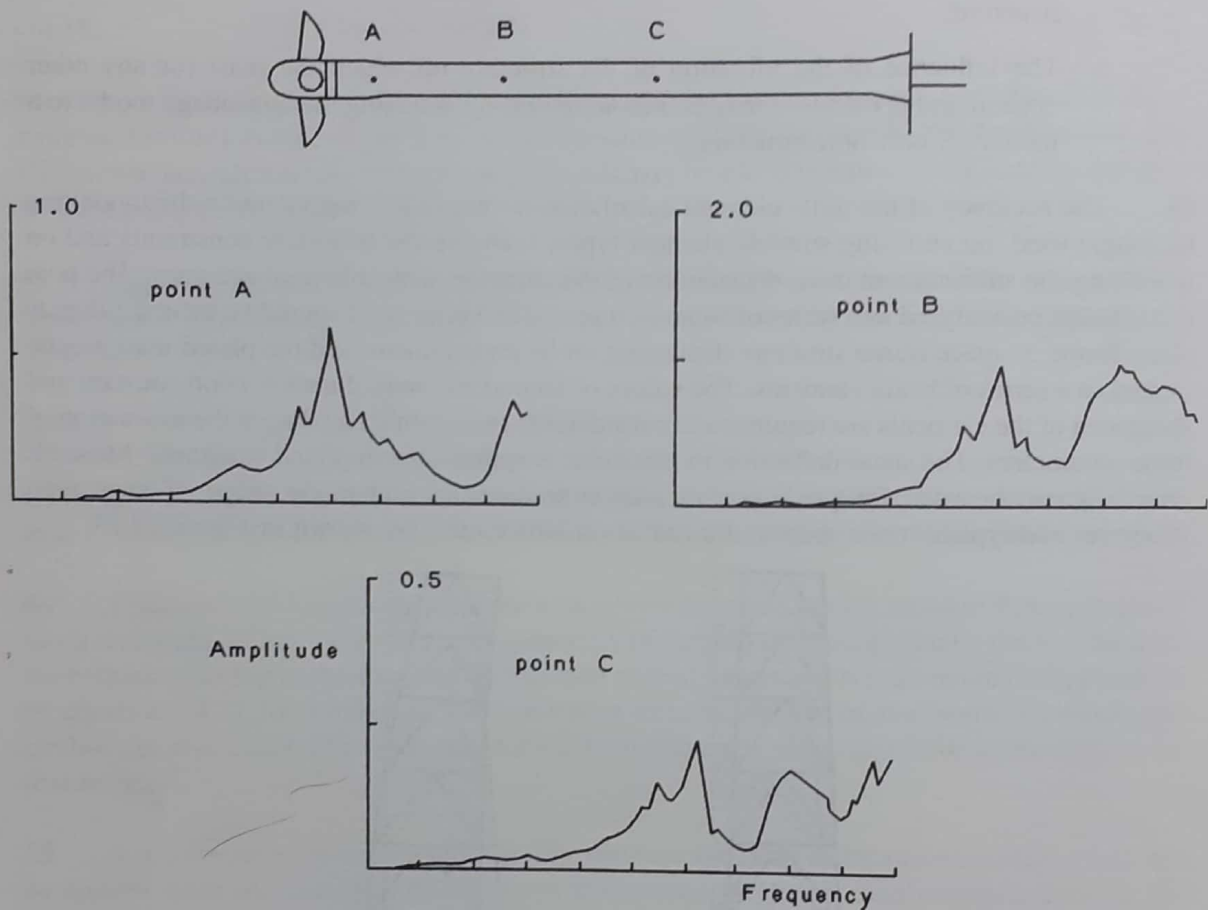


Figure 14.14 Vibration Response Characteristics of an A Bracket

90 A shaft bracket may be modelled as a simple plane frame structure using FE analysis for which it will be necessary to calculate correctly the sectional properties of both arms. A plane frame model of an A bracket and its corresponding mode shape are shown in *Figure 14.15*. Ideally a three dimensional FE model is necessary to represent the shaft and bracket combination but in practice the relatively low stiffness of the shaft compared with the bracket is such that a two dimensional representation is sufficient.

91 The treatment of shaft bossing vibration is similar to that for shaft brackets and is illustrated in the BSRA Ship Design Manual - Vibration (1981).

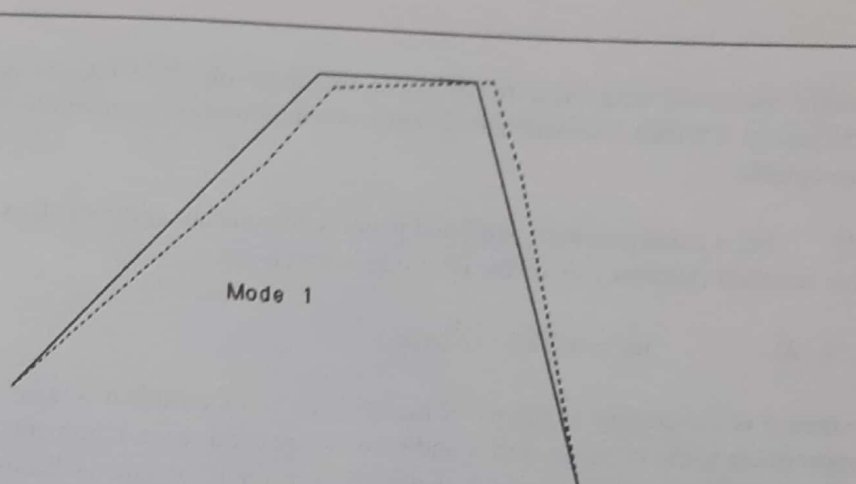


Figure 14.15 Plane Frame Model and Mode Shape of an A Bracket

Stiffened Panels

92 Local resonant vibration of individual plates and panels has been a source of concern to the ship designer and operator for many years. Many examples of the unacceptable vibration of deck panels, side and bottom plating and bulkheads have been recorded, some of which have been excited by the propeller and some by machinery, and which have caused fatigue and other failures. Panel vibration has also resulted in damage or malfunction of ship's equipment located nearby.

93 Whilst many qualitative reports have been published in the past, comprehensive quantitative knowledge of the subject is limited. Measurements carried out in recent years have indicated that all the published methods for predicting the natural frequencies of panels must be used with caution, since invariably they do not include, or in some cases are incapable of including, one or more of the important parameters necessary to provide accurate predictions, and in most cases are restricted to simple rectangular panels with ideal boundary conditions.

94 In principle the more complex finite element methods can overcome many of the problems of the empirically based methods but are rather expensive and cumbersome for use as a general design tool. Their use is therefore best confined to particularly problematic and sensitive areas such as complicated geometric structural arrangements close to exciting forces, areas supporting sensitive machinery, or large unsupported deck areas where vibration may cause degradation of equipment or personnel performance.

95 For panels which have either water or some other fluid on one or both sides it is necessary to consider the effect of the fluid loading on the vibration. The natural frequencies of a dry panel may be modified to account for the fluid loading which acts essentially as an added mass on the panel.

96 The approach advocated to avoid excessive vibration is to ensure that the individual panel natural frequencies do not coincide with, and in practice are more than 20% removed from, a

major excitation frequency, for example multiples of blade rate frequency or engine firing frequency. It is then assumed that the non-resonant vibration level will be sufficiently low as to be acceptable.

97 For a unidirectionally stiffened panel with constant stiffener sizes at constant spacing (s) the resonant frequency ω_r of the r^{th} mode is given by:

$$(14.18) \quad \omega_r = (\alpha_r^2/s^2) \sqrt{EIg/\rho A}$$

where ρ is the density of the panel material, α_r is a coefficient related to boundary conditions appropriate to the r^{th} mode, and A and I are the area and second moment of area about the neutral axis of a stiffener with an associated effective breadth of plating as discussed for stiffened panels under lateral load in Chapter 10. Values of α_r can be obtained from data sheet 18.91 based on Carmichael (1958) where it can be seen that for any mode from $r = 1$ to $r = 6$ there are a number of values α_r , each one associated with a value of a parameter ξ defining the degree of rotational support: ξ is zero for the simply supported case and infinite for clamped support. The choice of ξ must rely on the experience of the designer or advice should be obtained from specialists. Comparing the calculated natural frequencies of a stiffened deck panel, approximately $6.5 \text{ m} \times 4.5 \text{ m}$ with measured natural frequencies suggests that close agreement for this type of structure can be obtained by taking $\xi = 0$ and increasing the calculated frequency by 20%.

98 Equation 14.18 is valid provided that the transverse shear deformation of the beam is small, that the inertia effects due to rotation of each element of the beam negligible and that the plating carries its share of the bending loads indicated by the effective breadth chosen. Transverse shear and rotary inertia effects in beam vibration are discussed by Timoshenko and Young, 'Vibration Problems in Engineering', and are known to be of importance only in the higher modes for beams with a small span/depth ratio. Such proportions are rarely encountered in a ship and since only the lower modes of vibration of the plating are of interest to the ship structural designer, shear and rotary inertia effects may be neglected for conventional structures.

99 For a panel which is reinforced by a set of equal and equidistant girders in both directions as shown in *Figures 2.2 and 10.2*, the eigenvalue solution is obtained using orthotropic plate theory similar to that presented by Clarke (1977) for the buckling solution, and has been found to yield comparatively good results. The disadvantages of the method are:

- a. It yields good results only when the stiffener spacing is small compared to span.
- b. Torsion of the stiffening elements is neglected.
- c. For irregularly spaced stiffeners it is difficult to apply the theory.
- d. If the stiffener rigidity varies along the span it is difficult to apply the theory.

100 A detailed description of the method can be found in the BSRA Ship Design Manual - Vibration (1981). Based on this procedure, computer programs are available from various organisations including BMT CORTEC Ltd and Lloyd's Register of Shipping. The input to these

programs consists of plate geometry, regular and irregular stiffener properties, supporting arrangements and physical properties of the materials. A typical mode shape obtained from these programs is shown in Figure 14.16.

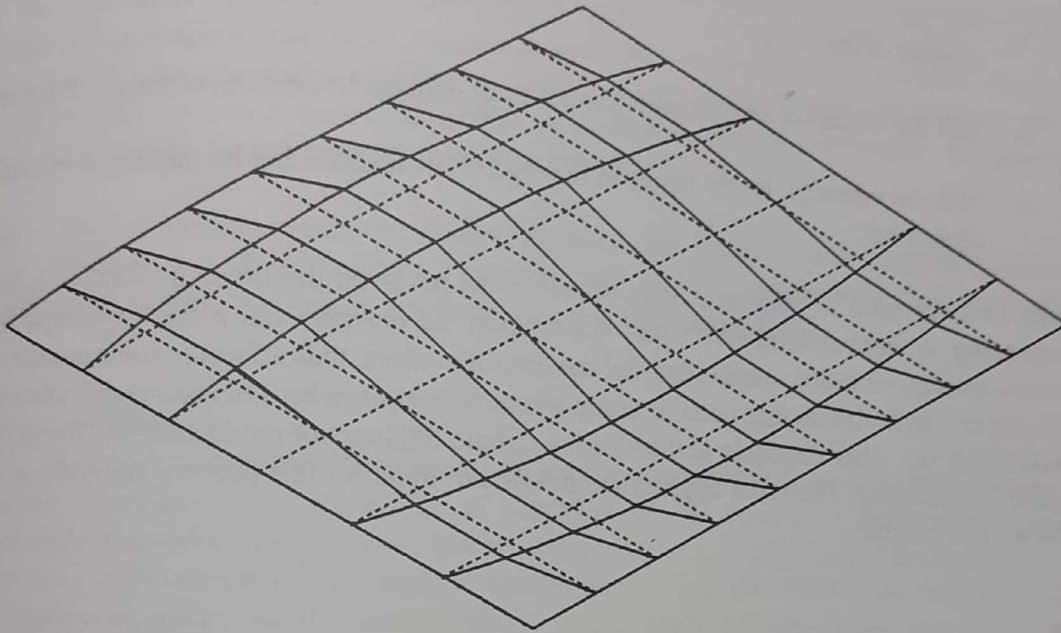


Figure 14.16 Typical Mode Shape of a Stiffened Panel

Equipment Damage and Malfunction

101 Any ship contains a large number of items of equipment, principally electrical or electronic equipment, which are sensitive to vibration. It is a requirement that equipment of this type undergoes vibration testing to ensure that it will withstand normal ship vibration. Nevertheless, failures do occur and the two main reasons for failure are:

- a. Large, heavy equipment is mounted on stiffened plate structure which has insufficient *dynamic stiffness*.
- b. Equipment is installed on anti-vibration or shock mountings which are selected on the assumption that they will be attached to a very stiff structure. In practice it is occasionally found that the supporting structure is less stiff than the mountings, in which case the equipment can be subjected to very high vibration levels.

It is important that the designer should examine the proposed sites of sensitive equipment and ensure that adequate supporting stiffness is provided. Generally the dynamic stiffness of the structure supporting the mountings should be at least 10 times the dynamic stiffness (in the same direction) of the mountings themselves.

Vibration Criteria and Limits

102 The three broad criteria by which the vibration characteristics of a ship may be judged are:

- a. Will the vibration levels have an undesirable effect on the working efficiency and/or comfort of the crew?
- b. Will the vibration levels result in structural damage in the form of fatigue fracture?
- c. Will the vibration levels result in damage to, or malfunction of, any items of ship's equipment?

In order to make such a judgement it is necessary to be given the limiting vibration values corresponding to these criteria. With this information the designer can make the comparison between estimated vibration amplitudes and the limiting values, thus affording some quantification of any design decisions that must be made where compromise between acceptable vibration characteristics and other desirable features, for example lightness or production-kindliness of a structure, is necessary. The most appropriate published limits for this purpose are given in the following paragraphs.

Working Efficiency and Personnel Discomfort

103 In 1984 the International Organisation for Standardisation (ISO) published International Standard ISO 6954-1984 (identical to BS6634 (1985)), 'Mechanical Vibration and Shock - Guidelines for the Overall Evaluation of Vibration in Merchant Ships'. It has been widely used to assess the acceptability of the vibration characteristics of newly-delivered ships and in recent years most new ship design specifications require that vibration levels will meet this standard.

104 An examination of ISO 6954-1984 or BS6634 shows that it does not, in fact, present a limiting value of vibration amplitude but a range of vibration amplitudes above which 'adverse comments are probable' and below which 'adverse comments are not probable'. It will also be noted, in the second paragraph of the document's introduction, that the standard 'reflects the current technology giving due regard to the overall acceptability judgements available based on technical as well as human performance or discomfort criteria'. These features of the document arise mainly from the fact that discomfort is a subjective experience which varies not only from person to person but will vary for any one person depending upon other environmental factors present, for example noise. However, despite its apparent imprecision, the standard has remained the most commonly accepted basis for the evaluation of ship vibration and although specific to merchant ships it is unlikely that any similar standard established for warships would present significantly different vibration amplitudes. Wide experience of the standard's use suggests that the upper boundary of the 'adverse comments not probable' zone should be regarded as the upper design limit since vibration amplitudes measured on the completed ship which significantly exceed this level will almost certainly require remedial action. It may, however, be prudent to specify lower vibration levels in compartments where delicate or sensitive work is undertaken, for example in operating theatres.

105 Another ISO document which, in the past, has been used occasionally to assess ship vibration is the draft international standard ISO DIS 2631/4, 'Guide for the Evaluation of Human Exposure to Whole Body Vibration - Part 4: Evaluation of Crew Exposure to Vibration Onboard Sea Going Ships (1 Hz - 80 Hz)'. This document was considered to have several deficiencies and has recently (1988) been withdrawn by ISO. It is not intended that it will be updated and reissued in the near future but the current (1989/90) 5-yearly review of ISO 6954-1984 will include consideration of appropriate information contained in it. The review will also address the problem of evaluating the cumulative effect of a number of separate vibration components and it is suggested that any new edition of ISO 6954, or its British Standard equivalent, will present the most reliable basis for the assessment of ship vibration.

106 Several national regulatory bodies, classification societies and research organisations have published ship vibration limits. However, with very few exceptions, they are in essence similar to ISO 6954-1984 but do not have the same wide acceptance.

Structural Damage

107 Although it is possible that structural damage could result from main hull vibration it is generally considered that amplitudes of vibration of the main hull girder which produce intolerable discomfort result in quite low structural stresses. It follows that limits for main hull vibration can be set from consideration of human comfort and ISO 6954-1984 can be considered an appropriate guide. Similar considerations would apply to some large part of the ship's structure, such as a deckhouse. However, stiffened deck panels, shell plate, small sections of structure or discrete items such as masts will not normally be associated with human comfort and in these cases the concern is the avoidance of fatigue cracks occurring in welds or thin plates. Unfortunately, because of the wide diversity of 'local structure' and the generally obscure relationship between the occurrence of a fatigue crack and the amplitude of vibration measured at (usually) some other point, it has not been possible to develop any reliable, definitive vibration limit for local structure. Experience gathered over a number of years has led to several marine organisations adopting their own guidance vibration levels. BMT CORTEC Ltd consider that local cracking may be associated with vibratory acceleration amplitudes of 5 m/s^2 (approximately 0.5 g). Det norske Veritas has suggested a maximum vibration velocity amplitude of 80 to 100 mm/s at the point of maximum vibration; at 9 Hz, in the middle of the frequency range of interest, this corresponds to 0.5 g.

108 It will be appreciated that these limits for the vibration of local structure are unlikely to be used in the context of vibration design since the prediction of a vibration amplitude, as opposed to the prediction of a natural frequency, will rarely be undertaken. They become of use only when attention is directed to the heavy vibration of some local structural area during sea trials.

Malfunction of Equipment

109 The available standards relating to the vibration of equipment are not directly of concern to the designer of the ship structure. The equipment manufacturers are required to carry out vibration tests on their equipment to ensure that it will withstand the ship vibration environment. In the case of Ministry of Defence ships, unless otherwise agreed, the equipment must be

subjected to the vibration tests detailed in Defence Standard 07-55, Part 2, Section 1. The levels of vibration at which the equipment is to be tested are specified according to which region in the ship the equipment is to be sited (for the purposes of that standard the ship is divided into three regions, the **masthead** region, the **after** region and the **main** region). For the designer, therefore, there is a clear implication that the main structure of the ship in these three regions must not exceed the corresponding vibration test amplitudes. Examination of test amplitudes indicates that they could be exceeded in the after and main regions if a main hull mode is excited into resonance and in the mast region if the fundamental mode of the mast is excited into resonance. If, however, the designer ensures that these phenomena will not occur at any common operational speed, the ship vibration environment will be less severe than that on which the Defence Standard is based. In addition, however, it is necessary to ensure that areas of local structure to which equipment is attached are sufficiently stiff to avoid a local amplification of the vibration of the main structure.

Specialised Ships

110 The previous paragraphs describe briefly the most commonly used vibration limits for ships. It is possible, however, that because of a ship's particular rôle, for example in research requiring a low underwater noise signature, the specification for the ship may call for more stringent limits than those quoted above. In such a case it is essential that the designer assesses the practicality, cost and other implications of the limits as early as possible in the design process.

Vibration Design Procedure

111 In order to ensure that a new ship will meet any specified vibration limits imposed, it is essential that the determination of the ship's inherent vibration characteristics should be recognised as an integral part of the overall ship design process. That is, vibration should be the subject of a design procedure which will continue throughout the ship's design and construction stages. In detail the procedure will depend upon the type of ship and on the designer's preferred method of working, but in every case it can be regarded as consisting of a number of stages as discussed in Chapters 1 and 6. In this publication they are termed initial design and first and subsequent iterations with, finally, contract definition. Only the first three stages are of principal relevance for design to resist vibration.

Initial Design

112 It can be seen from the foregoing paragraphs that early design decisions taken in relation to the principal factors of a new ship, for example speed, power, strength, operational rôle and general arrangement, will largely dictate its fundamental vibration behaviour in service. Consequently, it is at the stage when these decisions are made that the designer should investigate any major features which might introduce vibration problems. The process depends, to some

extent, on the designer's experience of what constitutes undesirable features in the context of vibration, but should include:

- a. Visual assessment of the general arrangement drawings, noting any special feature such as high masts or gantries, position of aerials, large unsupported deck areas and position of accommodation and working spaces.
- b. Assessment of machinery systems, noting magnitude and frequency of excitation sources, power/displacement ratio, type of machinery, propeller type (fixed pitch, controllable pitch etc) and propeller and hull configuration (see paragraphs 22 to 36).
- c. Assessment of the operational rôle, noting speed range, maximum and minimum displacements and requirement for any stringent vibration limits (see paragraph 110).
- d. Estimation of main hull natural frequencies using 'similar ship' data or graphs and comparison (in a qualitative sense) with excitation frequencies (see paragraphs 37 to 54).

The tasks outlined above can usually be completed fairly quickly but their execution is crucial in revealing vibration characteristics which, if ignored, could present major design problems at a later stage.

113 The first three of the four tasks rely heavily on accumulated experience and the structural designer will need to elicit the opinions of other specialists involved in, for example, propeller design and machinery arrangement. If the new ship incorporates relatively innovative features it may be helpful to seek the assistance of a vibration specialist.

114 The fourth task can be facilitated by use of a Campbell diagram which relates the main hull natural frequencies to the vibration excitation frequencies. The diagram will quickly provide a first indication of the extent to which the avoidance of main hull vibration presents a problem.

115 The Campbell diagram is a graphical presentation which has as its ordinate a frequency scale (usually 0 to 500 or 600 cycles/min) and as its abscissa a scale marked in revolutions per minute covering the whole range of main shaft speeds as shown typically in *Figure 14.17*. The calculated main hull natural frequencies appear on the diagram as horizontal lines since these frequencies do not change with ship's speed. The frequencies of the major vibration excitation forces usually correspond to integral multiples of main shaft speed. For example, propeller unbalance frequencies are equal to main shaft speed and blade rate is equal to the number of blades times main shaft speed. Consequently, they are represented on the diagram by straight lines having an appropriate slope. Clearly, where these lines intersect the main hull natural frequencies, resonance will occur at the corresponding main shaft speed. Generally several of the natural hull modes will be excited by one or other of the excitation sources at some main shaft speed. Whether or not these speeds are likely to be of consequence can be quickly determined from the diagram.

116 Figure 14.17 presents a Campbell diagram for a ship driven by a five bladed propeller at a maximum shaft speed of 250 rpm. If it is assumed that the propeller has controllable pitch and operates at a constant shaft speed of 250 rpm it can be seen that:

- Propeller blade rate does not excite any of the lower main hull modes.
- Propeller and shaft unbalance (shaft rate) will excite only the four node vertical (4NV) mode at a shaft speed of 269 rpm. This is higher than the maximum shaft speed but as the accuracy of the calculated 4NV frequency is probably not better than 10% it should be assumed that resonance will occur at 250 rpm and the amplitude of vibration should be calculated (see paragraphs 48-54). If the amplitude is significant, main hull vibration will be constantly noticeable if not excessive. This is clearly undesirable and can be avoided by changing either the proposed constant shaft speed or the number of propeller blades. Either change would, of course, imply modification of the propeller or machinery design and whilst this might be unwelcome it is probably only at the initial design stage that vibration can be given its due consideration in the selection of machinery.

117 If, on the other hand, the propeller has fixed pitch then vertical lines, indicating the principal operating speeds, should be added to the diagram. If the entire range of shaft speeds, that is 0 to 250 rpm, needs to be considered then it can be seen that:

- Propeller blade rate will excite all the lower hull modes at speeds below 112 rpm. However, calculation would probably show that because the blade rate force at 112 rpm is less than 10% of the force at 250 rpm (see paragraph 25) and damping

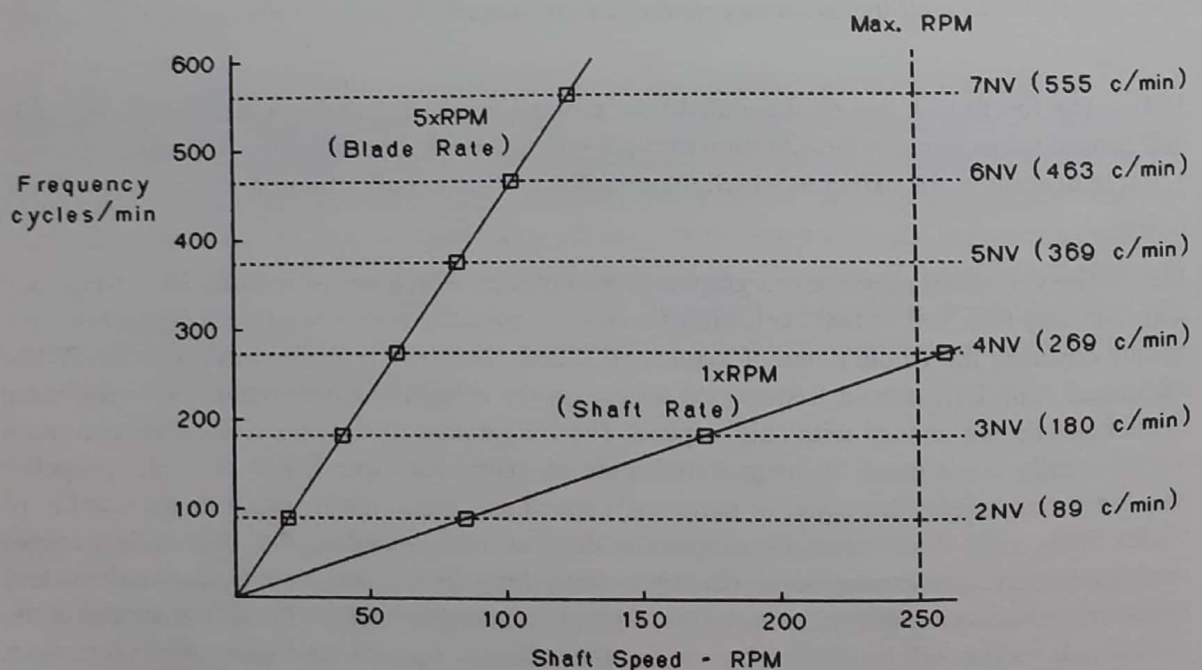


Figure 14.17 Campbell Diagram for Five Bladed Propeller

associated with the 5NV, 6NV and 7NV modes is high, the vibration amplitude will be low. Although the damping associated with 2NV, 3NV and 4NV modes is lower, the blade force below 54 rpm is less than 1% of the force at 250 rpm and again, calculated amplitudes should be acceptably low.

- b. The shaft rate can excite the 2NV, 3NV and 4NV at speeds of 89 rpm, 180 rpm and 269 rpm respectively. Vibration amplitudes should be calculated in each case.

118 A similar diagram should be produced for horizontal and torsional hull nodes. It is pointed out that, for the sake of clarity, Figure 14.17 shows only a single frequency for each mode. If a ship operates under different displacement conditions each mode will be associated with a range of frequencies bounded by those corresponding to the lowest and highest displacements. This range must be further extended to account for the inaccuracy involved in the calculated values, say about 10%. Consequently, each hull mode is represented by a horizontal band rather than a single line. In this circumstance the avoidance of exciting some main hull mode becomes difficult, if not impossible. The designer is faced with the task of assessing relative amplitudes, albeit qualitatively at this stage. Generally, the excitation at high powers of any major hull modes, usually 2NV to 7NV modes and 2NH to 5NH (horizontal) modes should be avoided, but even at low powers excitation of the lower modes, that is, two-node and three-node vertical and horizontal modes, can result in unacceptable vibration levels.

119 The examples given above are intended only to illustrate the purpose of this first step in the design procedure. Potential vibration problems will be revealed and the designer can note those which will have implications for later structural design and can bring to the attention of other members of the design team those features which have a considerable influence on vibration.

First Design Iteration

120 The work required during this stage consists largely of reviewing the results of the previous study in the light of more detailed information. Potential vibration problems revealed earlier can be quantified and design decisions influencing vibration made with more confidence. Much of the task is concerned with collating and correlating design details as they become available. The general pattern of the procedure at this stage is summarised as follows:

- a. **Vibration Specification** - decide any vibration limits specified for accommodation, working spaces, masts or sensors (see paragraphs 102 to 110).
- b. **Excitation** - obtain excitation frequencies and forces for the propulsion train, main machinery and other equipment, as well as from ship motion.
- c. **Main Hull Natural Frequencies** - estimate using most up-to-date information (paragraphs 40 to 47).
- d. **Main Hull Vibration Amplitudes** - calculate vibration amplitudes at resonance (see paragraphs 48 to 54) and blade rate forced vibration at maximum speed (see paragraphs 55 to 57).

- e. **Local Structure** - estimate resonant frequencies of masts (paragraphs 78 to 88), shaft brackets (paragraphs 89 to 91) and unsupported deck areas (paragraphs 91 to 100).
- f. **General Arrangement** - assess implications of relative positions of accommodation, working spaces and vibration sensitive equipment to excitation sources.
- g. **Comparison of Results** - assess the results of the above calculations in relation to specified vibration limits and note those areas in which vibration amplitudes might be considered excessive.
- h. **Solutions** - consider possible solutions to any potential vibration problems.

121 On completion of this work the principal vibration characteristics of the ship will have been established. The excitation of major modes of main hull vibration should have been avoided or calculated amplitudes at resonance shown to be acceptable. If this is not so, further attempts should be made to change the excitation frequencies or to reduce the magnitude of the vibratory forces. It is unlikely that the natural frequencies of the main hull can be changed significantly by structural modifications.

122 Problems associated with relatively large amplitudes of blade rate vibration in very high power/displacement ratio ships can be approached by placing accommodation and working spaces and sensitive equipment as far from the propeller as possible. Accommodation decks above the propeller are, in most ships, subject to vibration levels in excess of commonly specified vibration limits. Alternatively, or additionally, attention should be paid to the dynamic response of the aft end structure, that is the aftermost 10% of the ship length. The natural frequencies of deck panels, bulkheads, shell plating and other structure in this area should be well (>50%) above blade rate.

123 The calculated natural frequencies of other local structural areas should also be well above excitation frequencies. Where this is seen not to be the case the effect of structural modification should be explored.

Second and Subsequent Design Iterations

124 As a result of the work carried out during the two previous stages the designer should feel confident that any potential vibration problems associated with the basic design features of the ship, the principal dimensions, propulsive system, general arrangement and operation, have been avoided. The natural frequencies of local structural areas susceptible to vibration will have been calculated and those which give cause for concern noted for further attention. As the ship design becomes more detailed and crystallised it is necessary to confirm the absence of major problems and investigate in more detail any specific problems which have not yet been resolved. The main tasks can be summarised as:

- a. Confirm magnitude of propeller induced forces by reference to model tests and recalculate vibration amplitudes if necessary.

- b. Use analytical methods (paragraphs 99 to 100) or FEA methods (paragraphs 87 to 88, 90 and 94 to confirm natural frequencies of those local structural areas which gave cause for concern in the previous stage. Modify the structure if necessary and recalculate.
- c. Where the vibration of local structure is considered to be critical and may have a specified vibration limit, for example for masthead antennae, use FEA methods to calculate amplitude of vibration at the required operational speeds. If necessary, modify the structure, or supporting structure, and recalculate.
- d. Examine structures subjected to high repetitive impulsive forces, for example weapon platforms and, where considered appropriate, use FEA methods to calculate the dynamic response to the force (see paragraph 34). Modify structural design and if necessary recalculate.

125 In this stage of the design procedure attention is usually focussed on a small number of local structural areas. Calculation needs to be more rigorous and solutions are sought through changes in structural design details. However, the designer must take notice of, and assess any *changes* to the ship design or operational requirements, for example speed, power or propulsion system, which will have an influence on overall vibration characteristics.

Measurements During Construction and In Service

126 There are a number of uncertainties, assumptions and approximations associated with the prediction of ship vibration, particularly in the case of local structural areas, and the designer may be faced with the task of calculating the vibration response of some structure where errors are likely to be considerable. It is sometimes necessary or desirable to extend the design procedure to include the following tasks:

- a. Carry out vibration measurements on local structures where calculation is considered not sufficiently accurate, with a view to modifying the structure if necessary before outfitting.
- b. Carry out vibration measurements during acceptance trials to confirm the acceptability of vibration in the ship.

In addition to providing essential information on the current ship design, it will add considerably to the designers ability and confidence in undertaking vibration analysis of future designs.

Vibration Measurements and Troubleshooting

127 Whilst it is unlikely that many designers will be called upon to exercise those skills required in the measurement of vibration and the subsequent analysis and interpretation of the

measured values, it is to their advantage to be aware of the useful part that the results of a vibration experiment can play in the overall vibration design procedure. There are three stages in the procedure at which the designer will find that information derived from vibration measurements is essential or very desirable. These are:

- a. During the construction of the ship, vibration measurements in the part completed ship can be used to confirm design calculations and/or demonstrate the need for design modifications.
- b. On completion of the ship, the measurement of vibration during builders' trials provides the designer with feedback on the accuracy of his vibration design and should be considered as a contractual requirement on the shipbuilder.
- c. After the builders' trials or at some later time in the lifetime of the ship it may be necessary to provide a solution to some specific vibration problem.

The measurement procedures to be adopted in each of these three stages are briefly described below.

128 The measurement of vibration generally requires the attachment of a vibration transducer, that is a device which will convert the vibration acceleration or velocity, or displacement into an electrical signal, to the structure. (There are also 'non-interfering' vibration transducers using, for example, lasers or capacitance gauges which do not need to be attached to the structure, but these are rarely used in ship vibration work.) The electrical signals, which are usually very low and require some amplification are passed to either a measuring instrument capable of resolving a signal into its several frequency components and their associated amplitudes or, more commonly, are recorded on a paper chart or magnetic medium for subsequent analysis.

Tests During Construction

129 It will be found frequently that in calculating the natural frequencies of local structural areas or components it is necessary to make broad assumptions relating to their geometry and mass/stiffness distribution if unjustifiably extended and time consuming calculations are to be avoided. Where approximate calculations show that the natural frequencies lie in the region of excitation force frequencies (see paragraph 125) the designer may feel some lack of confidence in undertaking a design modification which compromises some other design aspect. It is advantageous, in this circumstance, to obtain support for a decision to alter the structure by subjecting the completed (or almost completed) structure to vibration tests.

130 The test is carried out by attaching several vibration transducers to the structure and inducing vibration of the structure by some artificial excitation force. There are two commonly used means of providing the force. The first is to use a single frequency sinusoidal force produced by two counter-rotating discs each having a large unbalanced mass; the frequency of the force is equal to the frequency of rotation of the discs, whilst the amplitude of the force is proportional to the square of the rotational speed. A device of this type is simple and robust and even relatively small ones (<50 kg) can provide a sinusoidal force of 10 kN at 1000 rpm which is enough to excite

even the whole hull to measurable amplitudes of vibration. Alternatively, a hydraulically operated exciter can be used which is more costly but has the advantage that it can deliver relatively high forces at very low frequencies. With either type of exciter the vibration test is carried out by sweeping through the frequency range of interest and measuring the induced vibration at sufficient points on the structure to obtain a well defined mode profile.

131 The second method of inducing vibration in the structure is to impart to it a series of hammer blows. The hammer used for this 'impact' method has a force transducer incorporated in its head so that the frequency and amplitude spectra of the force imparted to the structure with each blow are measured. The frequency and amplitude range will depend upon the magnitude of the impact and the material of the hammer head, for example steel, hard rubber or nylon. Ideally, the structure will respond by vibrating in all the modes which lie within the frequency band corresponding to the selected hammer head.

132 The impact method has the great advantage that the hammer is not attached to the structure, as in the case of the previously mentioned exciters, and the test can be carried out more quickly. In the majority of cases it is sufficient to identify the major natural frequencies of the structure and this method has been used very successfully for this purpose in investigating the vibration characteristics of masts, accommodation decks and even a large diesel main engine on its seating.

133 It is possible to extend the measurements, using either a mechanical exciter or hammer, to the determination of the structure's mobility (see paragraphs 71 et seq) but this requires a much more rigorous testing procedure and is useful only if the magnitude and the points of application of the forces to which the structure will be subject in service can be identified, for example for machinery seats. If underwater radiated noise is of concern, then each seating should be subjected to mobility measurements to ensure that no resonances in the seating coincide with any disturbing frequency of the supported equipment.

134 Clearly, the tests must be carried out at an appropriate stage in the construction of the ship. In general, that is when the structure is sufficiently advanced to be representative of the completed structure and whilst it is still possible to introduce structural modification. Vibration tests during construction have been widely used in the past to determine the natural frequencies of, for example, masts, shaft brackets, rudders, accommodation decks and generator flats.

Tests During Sea Trials

135 In the case of merchant ships it is now common practice to carry out both noise and vibration measurements during the shipbuilder's sea trials in order to provide a comparison between the limits stated in the Ship Specification and those existing in the completed ship. The basic vibration measurements can be performed with a convenient hand held instrument but, because vibration limits are specified in a large number of spaces, the tests can be time consuming. It is necessary, therefore, that a sufficient period of time (say four to five minutes per space) is allocated in the sea trials programme.

136 The most important feature of these tests is that they must correspond with the vibration requirements laid down in the Ship Specification. That is, they must be performed under the ship conditions specified, for example for maximum speed and cruising speed at defined positions in the ship. Where possible, the general conditions recommended in International Standard ISO 4867, 'Code for the Measurement and Reporting of Shipboard Vibration' (identical to BS6632 (1985)) for ship vibration trials should be observed. The results of the measurements must be presented in a form which allows direct comparison with the criteria or limits specified, for example acceleration, velocity or displacement, single amplitude or root mean square values, total level or the levels of the several frequency components (this last requirement implies the selection of a suitable bandwidth) so that agreement on the adequacy of the structure or achievement of criteria does not become confused.

137 Although not usually required of these trials, it is advantageous, should the opportunity occur, to extend the vibration measurements to provide some definition of the more global vibration characteristics of the ship including identification of the main hull natural frequencies, and amplitude of propeller or engine excited vibrations. The results of the measurements will provide useful data for future designs as an extension to the table in *Figure 18.86*.

Tests for Troubleshooting

138 Despite the best efforts of the designer, vibration problems may be identified when the ship first goes to sea, or original design features may be changed during the ship's life. It is not difficult for those serving in a ship to identify correctly that vibration is the cause of unacceptable discomfort, structural damage or equipment failure. However, the characteristics of the vibration, a knowledge of which is essential to the diagnosis of and resolution to the problem, cannot be quantified without appropriate measurements. Consequently, a report of a serious vibration problem almost always necessitates a vibration measurement trial.

139 The measurement of vibration for diagnostic purposes is no more difficult than that involved in the trials described above but the programme of measurements must be carefully planned. Some guidance on the necessary measurements might be obtained by careful scrutiny of the observations made by those on board but, generally, subjective impressions of vibration are indiscriminatory and can be misleading.

140 In most cases the trials programme consists of a series of experiments having the objective of determining the cause of unsatisfactory vibration excitation forces or dynamic structural responses (or both). As in most experiments, the general approach is to measure the change in vibration as the several controlling factors are changed, preferably one at a time. For example, the excitation force induced by a propeller can be changed by varying the shaft speed (frequency and magnitude) or by altering the propeller pitch (magnitude only) of a controllable pitch propeller. The dynamic response of a structure can be changed by modifying temporarily its stiffness or mass. For instance, the natural frequency of a generator flat might be increased by the use of temporary shores. Even the main hull natural frequencies may be modified by changing the variable load, for example between the fully loaded and the light condition.

141 Clearly the changes which can be introduced are limited and in some cases the effect of the change, though measurable, might be small. Consequently, the experimental changes must be well controlled and the results carefully annotated. Running a ship at sea is a costly operation and it is rarely that a ship can be taken to sea only for the purposes of carrying out vibration tests. It is imperative, therefore, that all the information needed for seeking a solution to the vibration problem should be gathered within the time allocated. Unlike many shore-based investigations, it is unlikely that a second opportunity will be made available. Hence the need for careful planning before the trial and meticulous execution of the test programme.

Interpretation of Vibration Measurements

142 Provided that the vibration trial has been carried out competently and the analysis and presentation of the information are clear, the interpretation of the results should not be difficult. Generally, the first objective is to determine at what position and under what conditions excessive vibration occurs (or in the case of tests carried out under construction, could occur). The second objective is to identify those factors contributing to the excessive vibration, either high excitation forces or conditions of resonance. In interpreting the results of vibration tests conducted in the ship whilst under construction it is the potential resonance conditions which are of immediate interest. The tests are intended to reveal the natural frequencies of the structure and to provide a good definition of the associated mode profiles. If any of the natural frequencies coincide with those of the major sources of vibration to be installed in the ship, the structure must be modified (see paragraphs 151 et seq). Because the structure under test might not be quite complete, the term 'coincidence' should include any natural frequency lying within the range of, say, 20% of an excitation frequency.

143 The interpretation of the results of vibration tests carried out during sea trials is more straightforward. In the first instance it is concerned only with identifying those positions at which the vibration level exceeds the required limit. If one or more positions are found to suffer from excessive vibration a troubleshooting trial may then be initiated since the original tests will allow only broad conclusions to be drawn.

144 The interpretation of troubleshooting investigations will often include the identification of natural frequencies and mode profiles as described above, but whereas vibration exciter measurements carried out in a 'quiet' ship allow precise measurement and control of the force frequency and magnitude applied to the structure, the vibratory forces applied to a ship *under way* can rarely be changed in frequency and magnitude in a controlled manner.

145 In some cases the vibration problem may be caused by inordinately large excitation forces and it is necessary to obtain measurements which will allow the assessment of the force characteristics. Generally, it is very difficult, though not always impossible, to measure directly the excitation forces induced by the propeller or main engine. However, it is usually possible to obtain useful information on the forces by measuring the change in vibration amplitude at the relevant frequency components, for example shaft or blade rate, at some non-resonant point such as the point of attachment of the shaft brackets to the hull as the shaft speed (or propeller pitch) is varied over a wide range. The blade rate vibration amplitude induced at some such point by a well designed propeller-hull configuration will be found to increase in approximately direct proportion

to the power or (rpm)³. If, at the higher shaft speeds or powers, excessive back cavitation develops on the propeller, the vibration amplitude will be found to be proportional to some higher power (7th, 8th or 9th) of the shaft speed. A graph plotting blade rate vibration amplitude against shaft rpm will show this large rate of increase at higher speeds. On a linear basis, however, the exponential increase in amplitude will be obscured by other features, such as hull resonances, but plotted on a log-log basis the resonances will be less prominent and any abrupt change in the slope of the curve, such as that brought about by the development of cavitation, will be easily discernible.

146 The above example illustrates the general approach to the interpretation of the results of many troubleshooting vibration trials. The vibration data are studied with the intention of quantifying the difference in behaviour between the excitation forces and dynamic response of the ship under tests and those associated with well designed machinery and structures. If this intention is fulfilled the problem can be diagnosed correctly and a solution sought.

Palliatives

147 Earlier in this chapter it was stated that the retrospective treatment of vibration problems is usually costly and might not always provide a complete solution. In all cases, reduction of the vibration amplitude of an existing structure can be achieved only by modifying the excitation force or by modifying the vibration response of the structure.

Modification of Excitation Force

148 Modification of the excitation force is usually the most effective, and in some cases the only, method of reaching a solution. For example, high levels of forced vibration induced over large areas at the aft end of a ship by the propeller. This type of vibration is usually associated with excessive propeller cavitation and can be treated satisfactorily only by reducing the propeller vibratory forces. The treatment would demand either a redesign of the propeller, an improvement of the flow into the propeller, or both. Consequently, it would be a task for hydrodynamicists and probably require extensive model testing. Though costly, a successful modification would not only remove the excessive vibration at the aft end but would reduce propeller excited vibration throughout the ship and noise in aft end accommodation areas.

149 A similar approach can be adopted in reducing vibration excited by large slow speed diesel engines. Their primary and secondary unbalance couples acting on the hull can be greatly reduced by the use of internal or external balancing devices. High torsional or axial vibration of the shafting system, inducing unacceptable vibration of the surrounding structure can often be modified by the installation of a torsional or axial damper.

150 It is clear that, although the modification of the vibration excitation force can result in a satisfactory reduction of the vibration over a wide area of the ship, it involves an extensive design effort and high installation costs. In the large majority of cases, therefore, the solution to a vibration problem is sought through modification of the dynamic response of the structure.

Modification of Structural Response

151 In principle, resonant vibration of the structure can be avoided by changing its mass or stiffness distribution. Additionally, the dynamic magnification at resonance (see paragraphs 15 to 17) can be reduced by increasing the damping associated with a structure. In practice, however, modification to damping is applicable only to light structures vibrating at relatively high frequencies. The natural frequency of a structure can be increased by reducing its mass or increasing its stiffness, and reduced by opposite changes. In most cases the design changes are directed to increasing the natural frequency so that it is appreciably higher than the frequencies of the major vibration excitation sources.

152 The application of the additional stiffening required to effect this change demands careful consideration. It will be noted that the natural frequency of a structure is associated with the square root of its stiffness (equation 14.4b). Hence the stiffness must be increased by approximately 20% in order to achieve an increase in the natural frequency of 10%. If the additional stiffening also contributes a significant additional mass even a 10% increase will not be realised, and it is therefore important to restrict extra stiffening material to the regions where it will be most effective. Changes in stiffness have most effect where the curvature of the mode shape is greatest while changes in mass have most effect where the displacement is greatest.

153 In the case of cantilevered structures, for example a mast or a cantilevered platform, the provision of adequate additional stiffening can be difficult and the possible contribution of other modifications such as the resiting of non-structural masses or the introduction of lightening holes should be considered. Alternatively, the increase in natural frequency can be sought by introducing restraints in the form of stays or braces, provided that a convenient strong and non-vibrating anchor point can be found. Where it is acceptable a further alternative is to reduce the length of the cantilever arm, that is reducing the height of a mast or the overhang length of a platform.

154 When faced with the task of increasing the natural frequency of this type of structure, it is important that the stiffness of the supporting structure should not be disregarded. For example, the natural frequency of a mast might be controlled predominantly by the stiffness of structure at its foot. Hence, additional stiffening of the mast can yield disappointing results. This situation can be avoided if the contribution made by the supporting structure to the resonant mode profile is determined by vibration measurements before modification.

155 The additional stiffening of a large (say more than 5 m square) stiffened deck panel introduces problems of a similar nature. It is frequently found that such a panel is resonating not in its fundamental mode but in some higher, more complex, mode. Consequently, the siting of additional stiffeners is fairly critical. Clearly, the addition of a girder along a nodal line will have little effect on the resonant mode, since the panel is virtually stationary along this line. The use of a pillar underneath, as a mid-span support, can be even more unrewarding. It will restrain the panel only at the point of attachment and has the added disadvantage that it provides efficient transmission of vibration from the deck below.

156 The most effective way of increasing the natural frequency of a large stiffened deck panel is to provide support in the form of a bulkhead (or partial bulkhead) below the deck. This is, of

course, a radical modification and it may not be attractive, or even feasible. In some cases, however, it has been found to be the only satisfactory solution.

157 On a number of occasions attempts to increase the natural frequency of some structural area by providing extra stiffness have been unsuccessful. The changes in dynamic stiffness introduced by additional steel are not easily judged, even by the practised eye. Any major modification can be undertaken with confidence only if the assistance of some of the methods described in this chapter is called upon. In particular, the use of vibration measurements to determine the natural frequency and mode profile of the structure before modification and the use of analytical methods, preferably a finite element method, to investigate the most effective way of changing the frequency so that it is well removed from the frequencies of the major vibratory forces, should be considered.